THE DESIGN OF A HIGHWAY FREIGHT TRAILER

12

A THESIS
Submitted in partial fulfillment
of the requirements for the Degree
of Master of Science in Mechanical Engineering

by

Frank Alfred Jones

Georgia School of Technology
Atlanta, Georgia
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INTRODUCTION

The design of highway freight trailers is influenced by many factors. It is the purpose of this thesis to investigate the principles controlling the design of trailers and to design a complete unit which can be manufactured with a minimum of materials and expense. It was endeavored to develop the information required for the maximum possible efficiency of design so that the trailer can be constructed to weigh as little as possible, commensurate with low cost of materials and manufacture. The information obtained includes data on certain materials which can not be used at the present time, due to war-time shortages, but which might be used in future designs.

The development of large, light-weight freight trailers has been made chiefly during the past ten years and there has been very little information published on this subject. Because of this, the various elements making up the trailer were taken up separately and studied in detail. Certain automotive components, such as the axle, wheels, rims, tires, landing gear, and springs, were not studied from the design standpoint but rather from the viewpoint of selecting from those available on the market. Since each of these items is manufactured by several firms devoted almost exclusively to producing particular items of these standard components, with a consequent realization of mass production economies, it is difficult to produce these items as cheaply as they can be purchased, unless such production is on a fairly large scale. The structural elements of the trailer were studied in detail so that they could be designed to be as light as possible without being too costly to produce and assemble. The design of these elements was taken up from a theoretical standpoint but it was necessary to temper the results of such studies with the knowledge of what will work under certain conditions, based on the writer's practical experience and observations, since the trailer is subject to indeterminate dynamic and impact loads.

The legal limitations of the states in which a trailer operates determine the size and payload of the trailer, so a study of these has been made first. Next to be taken up were the tools and materials available for fabricating the necessary parts. The selection of the standard automotive components then follows. After this the structural parts of the trailer are designed and the assembly and painting of the vehicle are discussed briefly. A description of the completed unit, along with weight and cost estimates, is then presented. Preloading for the prevention of fatigue failures is discussed, a guide compiled showing the procedure for normal maintenance, and the conclusions summarized.

LEGAL LIMITATIONS

The size of trailers is restricted by the various states, so these restrictions must be taken into account when determining the maximum over all size of the vehicle. Table 1 on page 6 shows the maximum width, height, and length allowed in the various states. The over all length shown is the combined length of the tractor and trailer combination. The emergency uniform standard shown on the bottom line is the standard set up by the Office of Defense Transportation as the maximum allowable sizes in all states for the duration of the War. This was done to enable highway transportation companies to move freight at maximum efficiency by running large size trailers through all states. In the past it has been necessary to transfer freight from long trailers to shorter ones at some state borders, since the maximum over all length allowed by some states is much less than that allowed by others. Since vehicles conforming to the uniform standards can operate in all states for the duration of the War and in a large number of states under the state restrictions, the sizes set up by this standard will be used for the purpose of designing the trailer.

The weight of trailers is restricted by the maximum weights allowed by the various states on one axle and on the tractor-trailer combination. These

TABLE 1. STATE RESTRICTIONS ON TRAILERS [1]

STATE	MAXIMUM DIMENSIONS				MAXIMUM WEIGHTS (LBS.)	
	WIDTH	HEIGHT	TRAILER LENGTH	OVER ALL LENGTH	ON ONE AXLE	TRACTOR & TRAILER
ALABAMA	96"	12'-6"	30'	40'	16,000	30,000
ARIZONA	96"	14'-6"	35'	65'	18,000	40,000
ARKANSAS	96"	12'-6"	35'	45'	16,000	40,200
CALIFORNIA	96"	13'-6"	35'	60'	18,000	800 (L + 40) [2]
COLORADO	96"	12'-6"	35'	40'	18,000	700 (L + 40)
CONNECTICUT	96"	12'-6"	40'	40'	*	40,000
DELAWARE	96"	12'-6"	33'	60'	18,000	40,000
DIST. OF COL.	96"	12'-6"	33'	33'	24,640	39,600
FLORIDA	96"	12'-6"	35'	45'	*	40,000
GEORGIA	96"	13'-6"	35'	45'	18,000	700 (L + 40)
IDAHO	96"	14'-0"	35'	45'	18,000	42,000
ILLINOIS	96"	*	35'	35'	16,000	40,000
INDIANA	96"	12'-0"	36'	40'	18,000	700 (L + 40)
IOWA	96"	12'-0"	33'	45'	16,000	450 (L + $53\frac{1}{3}$)
KANSAS	96"	12'-6"	35'	35'	18,000	700 (L + 40)
KENTUCKY	96"	11'-6"	26'-6"	30'	*	18,000
LOUISIANA	96"	12'-6"	33'	45'	18,000	20,000 NET LOAD
MAINE	96"	12'-6"	26'	40'	22,000	40,000
MARYLAND	96"	*	55'	55'	22,400	750 (L + 40)
MASSACHUSETTS	96"	*	33'	40'	30,000	40,000
MICHIGAN	96"	12'-6"	35'	50'	18,000	VARIES
MINNESOTA	96"	12'-6"	40'	40'	18,000	*
MISSISSIPPI	96"	12'-6"	40'	40'	18,000	30,000
MISSOURI	96"	12'-6"	33'	40'	16,000	38,000
MONTANA	96"	13'-6"	35'	60'	18,000	700 (L + 40)
NEBRASKA	96"	12'-0"	35'	42'	16,000	40,000
NEVADA	*	*	*	*	*	38,000
NEW HAMPSHIRE	96"	*	33'	45'	18,000	40,000
NEW JERSEY	96"	12'-6"	28'	45'	VARIES	60,000
NEW MEXICO	96"	12'-6"	35'	45'	18,000	600 (L + 40)
NEW YORK	96"	13'-0"	35'	50'	22,400	750 (L + 40)
NORTH CAROLINA	96"	12'-6"	35'	45'	18,000	40,000
NORTH DAKOTA	96"	12'-6"	35'	40'	18,000	40,000
OHIO	96"	12'-6"	35'	45'	18,000	30000 + (L × 750)
OKLAHOMA	96"	12'-6"	45'	45'	*	47,000
OREGON	96"	11'-0"	35'	50'	18,000	700 (L + 40)
PENNSYLVANIA	96"	12'-6"	33'	45'	18,000	39,000
RHODE ISLAND	102"	12'-6"	*	45'	22,400	40,000
SOUTH CAROLINA	96"	12'-6"	35'	45'	18,000	40,000
SOUTH DAKOTA	96"	13'-0"	30'	40'	*	30,000
TENNESSEE	96"	12'-0"	27'	35'	16,000	30,000
TEXAS	96"	12'-6"	35'	45'	18,000	700(L+40) MAX. = 38,000
UTAH	96"	14'-6"	45'	60'	18,000	700 (L + 40)
VERMONT	96"	12'-0"	50'	50'	*	40,000 ON STATE HIGHWAYS
VIRGINIA	96"	12'-6"	33'	45'	16,000	35,000
WASHINGTON	96"	12'-6"	35'	60'	18,000	700 (L + 40)
WEST VIRGINIA	96"	12'-6"	35'	45'	VARIES	VARIES
WISCONSIN	96"	12'-6"	33	45'	19,000	43,000
WYOMING	96"	12'-6"	40	45'	18,000	600 (L + 40)
EMERGENCY UNIFORM STD.	96"	12'-6"	35'	45'	18,000	40,000

* NOT RESTRICTED   [2] L = DISTANCE IN FEET BETWEEN FIRST & LAST AXLES

[1] THIS TABLE COMPILED FROM DATA PUBLISHED IN "STATE RESTRICTIONS ON MOTOR VEHICLE SIZES AND WEIGHTS - 1942 EDITION" BY NATIONAL HIGHWAY USERS CONFERENCE, PAGES 11 - 120

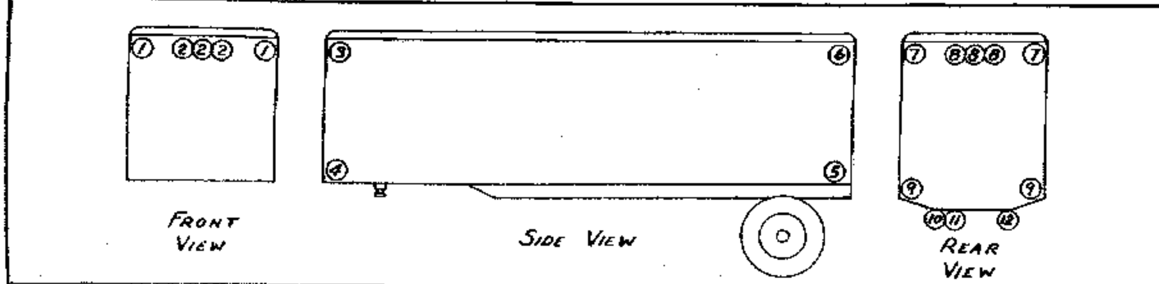
maximum weights are shown in Table 1. As in the case of the allowable sizes, emergency uniform standards have been set up by the Office of Defense Transportation and will be used in designing the trailer.

Table 2 on page 8 shows the lights and reflectors required for use on trailers operating in interstate commerce under the rulings and charter of the Interstate Commerce Commission. The lights required by certain states in addition to those required by the I.C.C. are shown by Table 3 on page 8.

The brakes used on trailers are subject to a number of requirements by the I.C.C.[1] Besides many requirements as to the quality of the individual parts used in the construction of the brakes, there are several very important requirements set forth in the I.C.C. brake provisions. It is required that the brakes operate in approximate synchronization with those on the towing vehicle, that the braking effort on the rearmost wheels shall be developed at the fastest rate (or that means be provided for applying the braking effort first on the rearmost wheels), that brakes shall operate on all wheels, and that application of the brakes on the trailer shall be made automatically in the case of a break-away. In the

1. See "Equipment Requirements for Motor Vehicles, 1942 Edition", published by the National Highway Users Conference, pages 7 - 10.

TABLE 2. LIGHTS AND REFLECTORS REQUIRED ON TRAILERS [1]



POSITION	TYPE OF LIGHT OR REFLECTOR	REQUIRED BY
1	AMBER CLEARANCE LIGHTS	INTERSTATE COMMERCE COMMISSION
2	AMBER IDENTIFICATION LIGHTS	CERTAIN STATES - SEE TABLE 3
3	AMBER CLEARANCE LIGHTS	I. C. C.
4	AMBER REFLECTORS	I. C. C.
5	RED REFLECTORS	I. C. C.
6	RED CLEARANCE LIGHTS	I. C. C.
7	RED CLEARANCE LIGHTS	I. C. C.
8	RED IDENTIFICATION LIGHTS	CERTAIN STATES - SEE TABLE 3
9	RED REFLECTORS	I. C. C.
10	RED TAIL LIGHT	I. C. C.
11	RED STOP LIGHT	I. C. C.
12	DIRECTIONAL SIGNAL LIGHT	CERTAIN STATES - SEE TABLE 3

TABLE 3. LIGHTS REQUIRED BY STATES IN ADDITION TO I.C.C. REQUIREMENTS [1,2]

STATE	ADDITIONAL LIGHTS REQUIRED			STATE	ADDITIONAL LIGHTS REQUIRED		
	3 AMBER TOP FRONT	3 RED TOP REAR	DIRECTIONAL BOTTOM REAR		3 AMBER TOP FRONT	3 RED TOP REAR	DIRECTIONAL BOTTOM REAR
ALABAMA	No	No	Yes	NEBRASKA	No	No	Yes
ARIZONA	No	No	Yes	NEVADA	No	No	No
ARKANSAS	Yes	Yes	Yes	NEW HAMPSHIRE	No	No	No
CALIFORNIA	No	No	Yes	NEW JERSEY	No	No	Yes
COLORADO	Yes	Yes	Yes	NEW MEXICO	No	No	Yes
CONNECTICUT	No	No	Yes	NEW YORK	No	No	Yes
DELAWARE	No	No	Yes	NORTH CAROLINA	No	No	Yes
DIST. OF COL.	No	No	No	NORTH DAKOTA	No	No	Yes
FLORIDA	No	No	Yes	OHIO	No	No	No
GEORGIA	No	No	No	OKLAHOMA	No	No	No
IDAHO	No	No	Yes	OREGON	No	No	Yes
ILLINOIS	Yes	Yes	Yes	PENNSYLVANIA	Yes	Yes	Yes
INDIANA	No	No	Yes	RHODE ISLAND	No	No	Yes
IOWA	Yes	Yes	Yes	SOUTH CAROLINA	Yes	Yes	Yes
KANSAS	Yes	Yes	No	SOUTH DAKOTA	Yes	Yes	Yes
KENTUCKY	No	No	No	TENNESSEE	No	No	No
LOUISIANA	Yes	Yes	Yes	TEXAS	No	No	No
MAINE	No	No	No	UTAH	No	No	Yes
MARYLAND	No	No	No	VERMONT	No	No	No
MASSACHUSETTS	No	No	Yes	VIRGINIA	No	No	Yes
MICHIGAN	No	No	Yes	WASHINGTON	No	No	Yes
MINNESOTA	Yes	Yes	Yes	WEST VIRGINIA	No	No	No
MISSISSIPPI	Yes	Yes	Yes	WISCONSIN	No	No	No
MISSOURI	No	No	Yes	WYOMING	No	No	Yes
MONTANA	No	No	No				

[1] COMPILED FROM DATA IN CATALOG No. 42 PUBLISHED BY R. E. DIETZ CO., NEW YORK, N.Y., PAGES 29-47.
[2] SEE "COMMERCIAL CAR JOURNAL," APRIL 1942, PAGE 29.

latter case, the application of the brakes must continue automatically for at least fifteen minutes. Since the majority of all trailers built operate in interstate commerce, the safety requirements of the I.C.C. should be met in every case.

AVAILABLE MACHINE TOOLS

The purchase of machine tools requires large investments of capital and such machines are very difficult to obtain during the present period of war-time scarcities, so the trailer will be designed so that only a very few such tools will be required. The structural parts of the trailer will be designed so that they can be completely fabricated from sheet steel and assembled with the following tools:-

1. Squaring shear having a capacity of 1/4" thick high-tensile steel 10 feet long.
2. Mechanical press brake having a capacity of 200 tons and equipped with square bend female dies, gooseneck male dies, narrow male die, 1/4" diameter beading dies, 3/8" diameter beading dies, and special die for forming ribs at one stroke.
3. Portable spot welder having full electronic control panels for synchronous welding control, having a capacity of 75 kva at 50% duty cycle, and equipped with a push gun with 12 foot dual cables and a pinch gun with 12 foot concentric cables.
4. Stationary dual-head series type spot welder having fully electronic control panels for

synchronous welding control and having a 10 foot platen length in the opening under the heads so that sheets ten feet wide by any length can be welded.

5. Angle rolls with proper rolls and enough capacity to roll 7" x 2" x 14 gauge angles leg-in and 3" x 2" x 12 gauge angles leg-out and 2" x 2" x 12 gauge angles leg-out.
6. Large size drill press with a capacity of 1 1/2" drill size, with spot-facing tools and tapping attachments.
7. Small size lathe with a swing of at least ten inches diameter by three feet long and equipped for cutting 1 1/2" National fine threads.
8. Miscellaneous portable hand tools including drills, air-driven wrenches and screwdrivers, grinders, disc sanders, and saws.
9. Arc welding machines with a rated capacity of 150 amperes.
10. Arc welding machines with a rated capacity of 300 amperes.
11. Oxy-acetylene cutting torches and contour cutting machine if possible.
12. Paint spray guns equipped with pressure feed tanks having air-driven agitators.
13. Air compressor with proper capacity to supply the compressed air needed by the machines and tools listed above.

AVAILABLE MATERIALS

The properties of various materials which might be used in the construction of the trailer are shown by Table 4 on page 13. Due to the present scarcities of stainless steels and the aluminum and magnesium alloys, the selection of the materials that will be used in the trailer will be confined to the carbon and high tensile steel groups. The data on stainless steels and the aluminum and magnesium alloys were presented so that the possibilities of their use might be studied when they once again become available for general use in industry. Although the alloying elements used in making high tensile steels are very scarce at the present, most steel companies secure the necessary amounts of these alloying elements by careful selection of the scrap steel used in making the high tensile steel. The use of high tensile steel in trailers at the present can be further justified by the weight of steel that will be saved if the superior properties of such steel are fully utilized. This fact has been recognized by the War Production Board in allotting alloy steel for use in trailers under its Controlled Materials Plan.

As every saving of a pound of weight in a trailer means that an extra pound of freight can be hauled and

TABLE 4. PROPERTIES OF VARIOUS MATERIALS WHEN COLD ROLLED INTO SHEETS [1]

MATERIAL	MANUFACTURER	TENSILE STRENGTH #/□"	YIELD POINT #/□"	SHEARING STRENGTH #/□"	% ELONG. IN 2"	% RED. OF AREA	HARDNESS BRIN.	HARDNESS ROCK.	IMPACT CHARPY FT. LBS.	IMPACT IZOD FT. LBS.	IMPACT TENSILE #/□"	END. LIMIT #/□"	MODULUS OF ELASTICITY	C	CR	MN	SI	P	CU	NI	MO	S	ZR	POUNDS PER CU. IN.
CARBON STEELS																								
S.A.E. 1010		56,000	33,000		35	65	113					28,000	29×10^6	.08-.13		.30-.50		.04 MAX.				.05 MAX.		.283
S.A.E. 1015		67,000	43,000		30	62	143					33,500	29×10^6	.13-.18		.30-.50		.04 MAX.				.05 MAX.		.283
S.A.E. 1020		69,000	48,000		30	63	143					34,500	29×10^6	.18-.23		.30-.50		.04 MAX.				.05 MAX.		.283
S.A.E. 1025		79,000	46,000		29	58	156					39,500	29×10^6	.22-.28		.30-.50		.04 MAX.				.05 MAX.		.283
HIGH TENSILE STEELS																								
ARMCO H.T.	AMER. ROLL. MILL	70,000	50,000		28-25	60		B72		130		45,000	29×10^6	.12		.20	.10	.05-.15	.35	.50	.05	.04 MAX.		.283
CORTEN	U. S. STEEL	70,000	50,000	59,000	27-22				40	60		45,000	29×10^6	.10	.75	.25	.75	.15	.40					.283
DYN-EL	ALAN WOOD	70,000	50,000-60,000		30-25		150			50	3,550	45,000	29×10^6	.12		.60		.085	.40			.03		.283
HI-STEEL	INLAND	70,000	55,000							65		49,000	29×10^6	.10		.60	.15	.12	1.10	.50				.283
MAYARI R.	BETHLEHEM	65,000-70,000	50,000-55,000	70,000-80,000	35-30	66	137			75		50,000	29×10^6	.08-.20	.20-1.00	.50-1.00	.05-.50	.04-.12	.50-.70		.05			.283
N-A-X 9115	GREAT LAKES	75,000	50,000-55,000		23	72	150			50		46,000	29×10^6	.10-.20	.50-.66	.50-.75	.60-.90	.04			.05		.10-.20	.283
RDS GRADE I	REPUBLIC	75,000	60,000		25					76			29×10^6	.12		.50-1.00		.04	.50-1.50	.40-.80	.20	.04		.283
YOLOY No. 908	YOUNGSTOWN	70,000	55,000	50,000	28	58	160		43			45,000	29×10^6	.06-.11		.30-.60	.10-.20	.05	.85-1.05	1.50-2.00		.05		.283
STAINLESS STEELS																								
TYPE 301		90,000	45,000		55		140-170	B80-B85					29×10^6	.08-.20	16.-18.	2.00 MAX.	.03 MAX.			6.0-8.0		.03 MAX.		.285
TYPE 302		80,000-95,000	30,000-45,000		55-60	60-75	130-175	B75-B88		100			29×10^6	.08-.20	17.-19.	2.00 MAX.	.03 MAX.			8.0-10.0		.03 MAX.		.286
TYPE 304		80,000-90,000	40,000		60-65	70	125-150	B70-B80		100		51,000	29×10^6	.08 MAX.	18.-20.	2.00 MAX.	.03 MAX.			8.0-10.0		.03 MAX.		.286
TYPE 305		90,000	45,000		55	70	130-162	B76-B85		100			29×10^6	.20 MAX.	19.		.03 MAX.			9.		.03 MAX.		.286
TYPE 306		85,000	40,000		60		125-150	B70-B80		100			29×10^6	.08 MAX.	19.		.03 MAX.			9.		.03 MAX.		.286
ALUMINUM ALLOYS														CU	SI	MN	MG	ZN	NI	CR	REMAINDER = AL			
17S-0	ALUM. CO. OF AMER.	26,000	10,000	18,000	20		45					11,000	10.3×10^6	4.0		.5	.5							.101
17S-T	ALUM. CO. OF AMER.	60,000	37,000	35,000	20		100					15,000	10.3×10^6	4.0		.5	.5							.101
24S-0	ALUM. CO. OF AMER.	26,000	10,000	18,000	20		42						10.3×10^6	4.2		.5	1.5							.100
24S-T	ALUM. CO. OF AMER.	68,000	44,000	41,000	19		105					16,000	10.3×10^6	4.2		.5	1.5							.100
53S-0	ALUM. CO. OF AMER.	16,000	7,000	11,000	25		26					7,500	10.3×10^6		.7		1.25			.25				.097
53S-T	ALUM. CO. OF AMER.	39,000	33,000	24,000	14		80					11,000	10.3×10^6		.7		1.25			.25				.097
MAGNESIUM ALLOYS														MG	MN	ZN	AL							
AM3S-0	AMER. MG. CORP.	32,000	17,000	19,000	16		44					8,000	2.4×10^6	98.5	1.5									.064-.067
AM-C52S-0	AMER. MG. CORP.	38,000	25,000	21,000	18		54					11,000	2.4×10^6	95.8	.2	1.0	3.0							.064-.067

[1] COMPILED FROM INFORMATION IN "METALS AND ALLOYS DATA BOOK," BY S. L. HOYT AND INFORMATION FURNISHED BY THE MANUFACTURERS LISTED ABOVE.

since the high tensile steels are superior to the carbon steels in every respect, high tensile steel will be used almost exclusively in the trailer. This will result in a slightly higher cost due to the higher cost of high tensile steel, but this small extra cost will enable the trailer to be made so much lighter that it will be easier to sell at a higher price than if it were cheaper and heavier. Users of trailers are very weight-conscious and have always been willing to pay considerably more for a light weight trailer than for a heavier one.

The properties of easily secured floor materials for the trailer are shown by Table 5 on page 15. Examination of the table will show that both shortleaf and longleaf pine have a strength very close to that of white oak. However the white oak is far superior in resistance to splitting and is far harder than either of the pines. There is no practical measure of the resistance to splitting, but a great deal of observation of trailer floors by the writer has revealed that white oak is far superior to other woods in resistance to splitting and abrasion. The hardness is a measure of the resistance to splitting and scuffing and the table shows that white oak is the best material in this respect. In consideration of the above, white oak has been selected as the floor material for the trailer and the design of the floor structure will be based on this.

TABLE 5. PROPERTIES OF VARIOUS WOODS [1]

NAME OF WOOD	MOISTURE CONTENT	WEIGHT PER BOARD FOOT [2]	STRENGTH				
			STATIC	IMPACT			HARDNESS
	%	LBS.	MODULUS OF RUPTURE IN LBS. PER SQ. IN.	FIBER STRESS AT PROPORTIONAL LIMIT LBS./SQ.IN.	HGT. OF DROP 50# HAMMER TO CAUSE COMPLETE FAILURE	IMPACT STRENGTH-WEIGHT RATIO [3]	LOAD REQ. TO EMBED .444" DIA. BALL TO ½ ITS DIA.
OAK, WHITE	12	3.644	13,900.	17,400.	39"	10.7	1,330.#
PINE, SHORTLEAF YELLOW	12	3.000	12,800.	13,600.	33"	11.0	690.#
PINE, LONGLEAF YELLOW	12	3.420	14,700.	15,400.	34"	9.94	870.#
FIR, COMMERCIAL WHITE	12	2.250	9,300.	11,200.	20"	8.89	460.#
POPLAR, YELLOW	12	2.330	9,200.	13,500.	20"	8.59	450.#

1 COMPILED FROM DATA IN "WOOD HANDBOOK," BY FOREST PRODUCTS LABORATORY, UNITED STATES DEPARTMENT OF AGRICULTURE, PAGES 46-53.

2 WEIGHT OF OAK WAS TAKEN FROM STOCK ON HAND.

3 CALCULATED BY DIVIDING VALUE IN COLUMN 6 BY VALUE IN COLUMN 3.

WEIGHT DISTRIBUTION

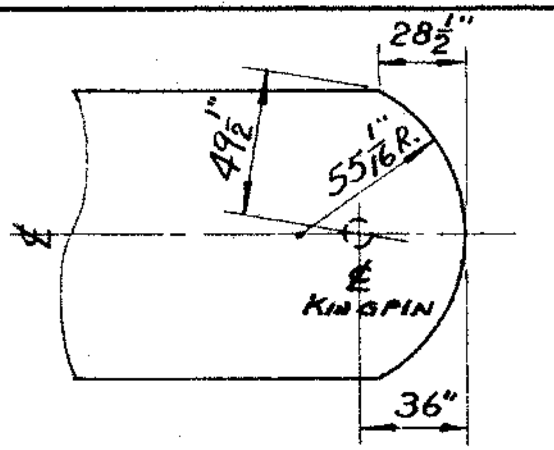
The distribution of the weight of the payload in the trailer is determined by the location of the kingpin and the location of the axle. The kingpin should be located as far back as possible to secure proper loading on the rear tires of the tractor while at the same time maintaining proper clearance between the front of the trailer and the rear of the tractor cab. When the tractor is turned to nearly right angles to the trailer, the condition of minimum clearance results. This condition is obtained only when making very sharp turns or in backing, both of which must be done at low speeds. When the trailer is traveling at high speeds there should be more clearance allowed between the tractor and trailer, as interference between them under such conditions would result in serious trouble. Table 6 on page 17 shows contours of the front end of the trailer and kingpin locations which have been found to work with practically all makes of tractors and under all normal conditions. The chart shows the contour used on both the round nose trailer and the square nose trailer with rounded corners (commonly called bull nose trailer).

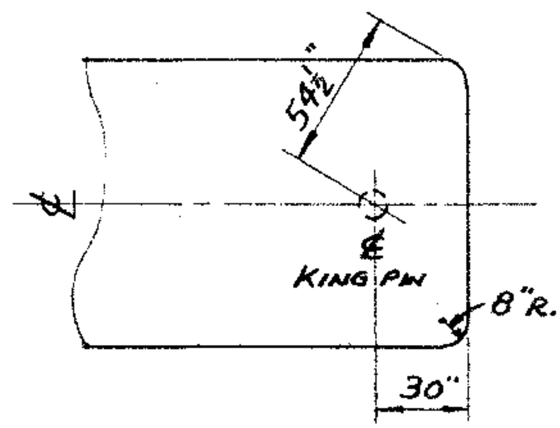
Table 6 also shows the probable actual distribution of the payload in the front of the two types of trailers.

TABLE 6. LOAD ON FRONT END OF TRAILER

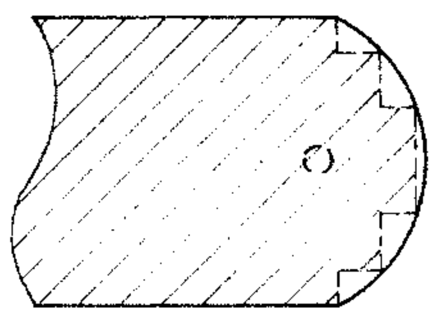
ROUND NOSE TRAILER	SQUARE NOSE TRAILER WITH ROUNDED CORNERS

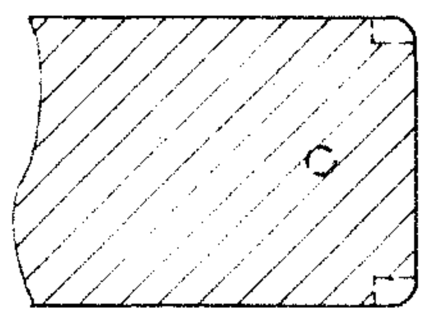
DIMENSIONS OF FRONT END OF TRAILER



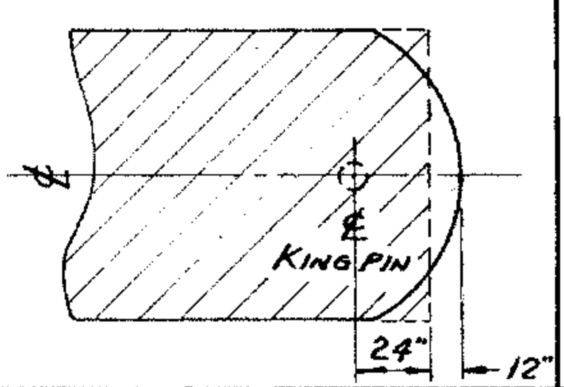


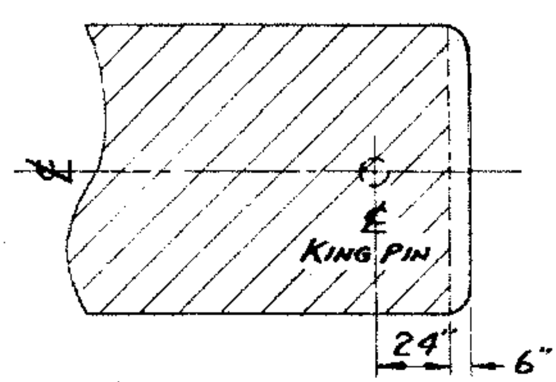
ACTUAL LOAD DISTRIBUTION





ASSUMED LOAD DISTRIBUTION





This is based on the fact that the majority of normal freight cargo is shipped in rectangular containers or crates. With this assumption in mind, the load distribution in the nose of the trailer is arbitrarily taken as that shown in the bottom section of Table 6. This is for the purpose of calculating weight distribution and resultant stresses.

When a trailer is loaded the cargo is packed tightly against the nose of the trailer and the loading is then continued until it reaches as far back in the trailer as possible. It is only rarely that the cargo is of such dimensions that the trailer can be loaded completely to the back end of the trailer and this results in considerable space being left at the rear end. For this reason the cargo will be considered as extending back only to within two feet of the rear end of the trailer. As in the case of the assumed load distribution in the nose of the trailer, this assumption is made for the purposes of calculations only.

The location of the centerline of the axle from the rear end of the trailer is shown by Table 7 on page 19. This is not based entirely on weight distribution, but has been worked out with the idea of keeping the wheelbase as small as possible while achieving the best appearance to the eye and good distribution of the weight of the payload. In the case of the 32 foot trailer, the distance from the

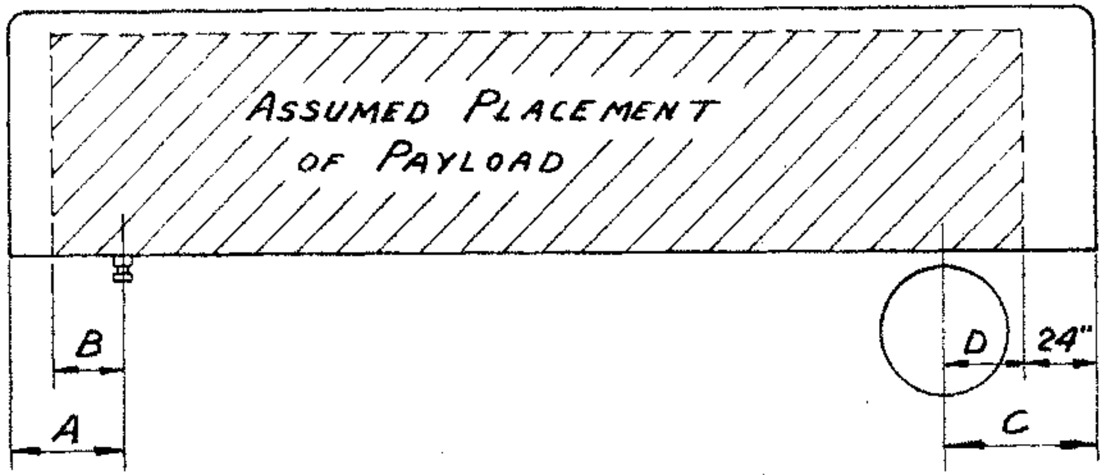
TABLE 7. LOCATION OF AXLE.

LENGTH OF TRAILER	DISTANCE FROM REAR END TO CENTER LINE OF AXLE
32 FT.	57"
30 FT.	50"
28 FT.	50"
26 FT.	43"
24 FT.	43"
22 FT.	43"
20 FT.	43"

rear to the axle has been increased to 57 inches in order to keep the length of the stressed members of the trailer as short as possible as well as to make the wheelbase as short as possible. In the case of the shorter trailers, the distance of 43 inches from the rear end to the axle is as small as it can be made and still allow enough room for the spring assembly to mount on the trailer. The axle locations selected have been tried on many trailers designed by the writer and are the result of successive changes in these locations due to the practical experience gained from actual operation of the vehicles. The locations of the axle shown in Table 7 give very good maneuverability and give a balanced appearance to the eye.

The weight distribution of the payload resulting from the axle and kingpin locations selected are shown by Table 8 on page 21, based on a payload of 30,000 pounds. This load is shown in both pounds and percentage of the total payload of 30,000 pounds. This payload figure was determined by taking a maximum gross weight of 45,000 pounds as the estimated weight of the trailer and 8,000 pounds as the estimated weight of the tractor. The smaller size tractors will approximate this weight of 8,000 pounds but the larger sizes weigh considerably more, depending on the horsepower and whether the engine is a gasoline or Diesel type.

TABLE 8. DISTRIBUTION OF PAYLOAD.



TRAILER LENGTH	A		B	C	D	PAYLOAD	LOAD AT KINGPIN		LOAD AT AXLE	
	ROUND NOSE	BULL NOSE				LBS.	LBS.	%	LBS.	%
32 FT.	36"	30"	24"	57"	33"	30,000.	13,430.	44.8	16,570.	55.2
30 FT.	36"	30"	24"	50"	26"	30,000.	14,890.	49.6	15,110.	50.4
28 FT.	36"	30"	24"	50"	26"	30,000.	14,890.	49.6	15,110.	50.4
26 FT.	36"	30"	24"	43"	19"	30,000.	15,320.	51.1	14,680.	48.9
24 FT.	36"	30"	24"	43"	19"	30,000.	15,320.	51.1	14,680.	48.9
22 FT.	36"	30"	24"	43"	19"	30,000.	15,320.	51.1	14,680.	48.9
20 FT.	36"	30"	24"	43"	19"	30,000.	15,320	51.1	14,680.	48.9

The load at the kingpin and the load at the axle were calculated by writing moment equations and solving by substitution. The following example is for the case of the 30 foot round nose trailer:

X = load at kingpin
Y = load at axle
360" = length of trailer.
36" = distance from front of trailer to kingpin.
24" = distance from kingpin to front end of assumed load.
50" = distance from axle to rear end of trailer.
26" = distance from axle to rear end of assumed load.

Equation (1)

$$X + Y = 30{,}000$$
$$Y = 30{,}000 - X$$

Equation (2)

$$+30{,}000 \left(\frac{324}{2}\right) - 24\,X - 298\,Y = 0$$
$$4{,}860{,}000 - 24\,X - 298\,(30{,}000 - X) = 0$$
$$274\,X = 4{,}080{,}000$$
$$X = 14{,}890 \text{ lbs.}$$

Equation (1)

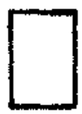
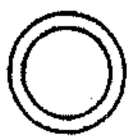
$$Y = 30{,}000 - X$$
$$Y = 15{,}110 \text{ lbs.}$$

SELECTION OF AXLE

The axle of a trailer consists mainly of a one piece member turned or forged down at the ends and machined for installation of tapered roller bearings. A flange is welded or forged on each side so that the brake drum can be attached to the flange by riveting or bolting. The four principal types of axles are compared by Table 9 on page 24. Examination of this table will show that the tubular type axle is superior to all the others with the one exception that it is slightly heavier than the I beam type. This slight difference in weight is compensated for by the facts that the axle saddle part of the spring assembly is heavier on the I beam type and that the I beam type requires a special piece where the U bolts fit around the axle, whereas the tubular type permits the U bolts to be formed so that they fit around the axle itself without needing any extra pieces. Since the tubular axle is far superior to the I beam axle in all respects except weight and since the weight difference is offset by the factors listed above, the tubular axle has been selected for the trailer.

The weights and costs of tubular axles having a rated capacity of 17,000 pounds are shown by Table 10 on page 25. The air brake and vacuum brake types of axles are compared, although the type of brake to be used will depend entirely on that being used in the prospective customer's

TABLE 9. COMPARISON OF AXLES (17,000 LBS. CAPACITY).[1]

TYPE AXLE	CROSS SECTION	SECTION DIMENSIONS	INDEX OF STIFFNESS			WGT. OF 5 FT. LGTH.
			VERTICAL	HORIZONTAL	TORSIONAL	
SQUARE		$3\frac{1}{2}'' \times 3\frac{1}{2}''$	12.50	12.50	25.0	208 LBS.
RECTANGULAR		$2\frac{3}{4}'' \times 4\frac{1}{4}''$	17.60	7.37	24.40	163 LBS.
I BEAM		$2\frac{3}{4}'' \times 4\frac{1}{4}''$	13.40	3.28	* [2]	109 LBS.
TUBULAR		$5'' \times \frac{1}{2}''$ TK.	18.10	18.10	36.20	120 LBS.

[1] COMPILED FROM DATA PUBLISHED IN 1935 BY THE TIMKEN-DETROIT AXLE CO.
[2] NOT SHOWN AS THE TORSIONAL STIFFNESS CAN NOT BE ACCURATELY CALCULATED.

TABLE 10. WEIGHTS[1] AND COSTS[2] OF AXLES (17,000 LBS. CAPACITY).

TYPE OF AXLE	MANUFACTURER	TYPE OF BRAKES	AXLE ONLY[3]		BRAKE KIT		TOTAL	
			WGT.	COST	WGT.	COST	WGT.	COST
TUBULAR	TIMKEN-DETROIT	AIR	429#	$146.26	50#	$29.60	479#	$175.86
TUBULAR	TIMKEN-DETROIT	VACUUM	399#	$119.25	75#	$24.80	474#	$144.05

1 ACTUAL SCALE WEIGHTS.

2 COSTS DELIVERED TO BRISTOL, VIRGINIA VIA LESS-CARLOAD FREIGHT.

3 BRAKE DIAPHRAMS ARE INCLUDED WITH AXLE ON AIR BRAKE AXLE ONLY.

fleet of vehicles, since practically all freight line operators will switch trailers from one tractor to another and thus will require the same type of brakes on all their vehicles. The table shows that there is very little weight difference between the air and vacuum axles complete with brake kits, but that the air brake equipment is quite a bit more expensive than the vacuum type. The axle having a rated capacity of 17,000 pounds was chosen because it provides approximately the required capacity for the payload and sprung body weight carried by the axle.

SELECTION OF WHEELS, TIRES, AND RIMS

The wheels used on trailers are of two principal types, spoke and disc. The spoke wheels are usually made of malleable iron and include the hubs and brake drums as integral parts of the wheel while the spoke wheels are pressed from steel plate and have the rim attached permanently to the wheel, but require separate hubs and brake drums. On the other hand, the spoke wheels do not include rims but do include spacer rings for maintaining the proper dual spacing of the wheels.

A comparison of spoke wheels and disc wheels is shown by Table 11 on page 28. Table 12 on the same page shows the hubs and brake drums which must be used in conjunction with the disc wheels. It is apparent that the selection of the wheels can not be made until all the parts such as hubs and drums and rims are considered, both from the weight and cost standpoints. The selection will therefore be left until all these other items can be considered at one time.

The larger sizes of trailer tires are compared in Table 13 on page 29. Since the weights and costs of the tires affect the selection of the wheels, the selection of all the component parts of the wheel assembly will be made by comparison of all the possible combinations.

TABLE 11. COMPARISON OF WHEELS

TYPE	MANUFACTURER	SIZE	WEIGHT	COST [1]	REQUIRED FOR ONE AXLE		
					NO.	WGT.	COST [1]
SPOKE [2]	ERIE	20"	185 #	$28.45	2	370#	$56.90
SPOKE [2]	ERIE	22"	198 #	$31.75	2	396#	$63.50
DISC [3]	BUDD	20"	105 #	$16.28	4	420#	$65.12
DISC [3]	BUDD	22"	117 #	$17.50	4	468#	$70.00

[1] COST DELIVERED TO BRISTOL, VIRGINIA VIA LESS-CARLOAD FREIGHT.

[2] SPOKE WHEELS INCLUDE HUBS, BRAKE DRUMS, AND RIM SPACER RINGS.

[3] DISC WHEELS INCLUDE RIMS BUT DO NOT INCLUDE HUBS AND BRAKE DRUMS.

TABLE 12. WEIGHT AND COST OF HUBS AND BRAKE DRUMS [1]

MANUFACTURER	AXLE SIZE	WEIGHT	COST [2]	REQUIRED FOR ONE AXLE		
				NO.	WGT.	COST
TIMKEN-DETROIT	17,000 LBS. CAP.	150 #	$29.53	2	300#	$59.06

[1] USED WITH DISC WHEELS. [2] COST DELIVERED TO BRISTOL, VA. VIA L.C.L. FREIGHT.

TABLE 13. COMPARISON OF TIRES [1]

SIZE OF TIRES	RATED CAPACITY	WEIGHT OF TIRE, TUBE, AND FLAP	DIAMETER OF TIRE	REVOLUTIONS PER MILE	COST[2] (NOT INCLUDING 5% DEDUCTIBLE TAX)
10.00 × 20	4,000 LBS.	123.9 LBS.	41.12"	512	$54.99
10.00 × 22	4,275 LBS.	136.2 LBS.	43.19"	486	$58.35
11.00 × 20	4,500 LBS.	147.8 LBS.	42.78"	497	$61.85
11.00 × 22	4,750 LBS.	158.2 LBS.	44.80"	473	$80.20

[1] COMPILED FROM DATA IN "GOODRICH TIRE DATA BOOK," SHEET B-3, DATED JULY 1943.

[2] DELIVERED COST AS OF MARCH 1, 1944.

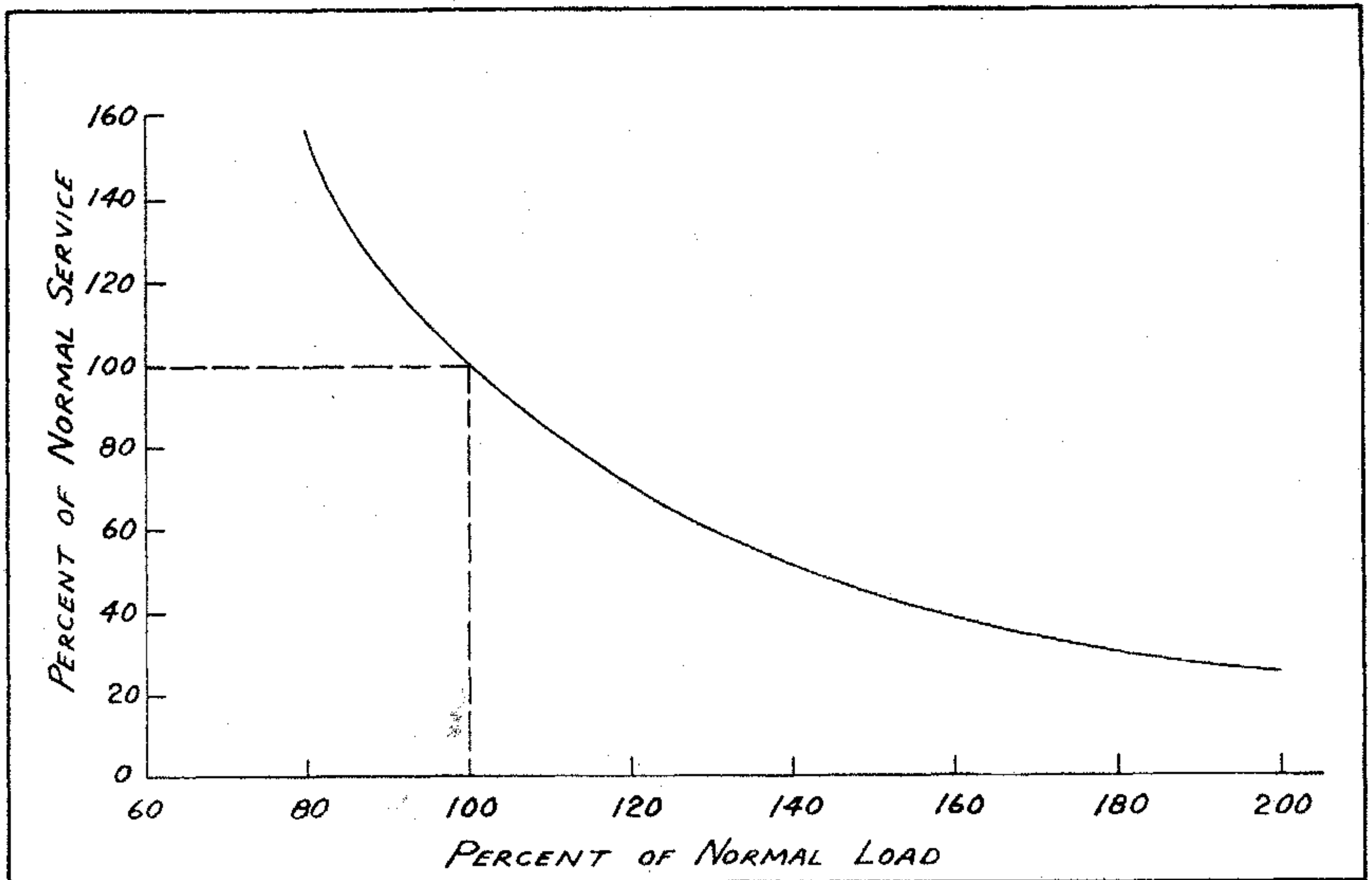
The effect of load on tire mileage is shown by Table 14 on page 31. This chart shows that with 80% of normal (rated) load on the tires the life (mileage) will be about 150% of normal while with 20% overload the life of the tire will be only about 70% of normal. It is apparent from this that it will be most economical to operate the tires in a slightly underloaded condition. An additional reason to refrain from overloading tires is that blowouts are often caused by overloading, resulting in destruction of the tire and tube and sometimes in the wrecking of the vehicle.

The proper rim sizes for trailer tires is shown by Table 15 on page 32. These are the wide base rim standards as recommended by the National Wheel and Rim Association for maximum mileage.

A comparison of the complete wheel assemblies is made in Table 16 shown on page 33. Examination of this table shows that the spoke wheel assemblies are both cheaper and lighter than the disc wheels in every tire size combination. For these reasons the spoke wheel assembly will be chosen for the trailer.

Table 17 on page 35 shows the weights and costs of spoke wheel assemblies including tires. The figures given are for the quantities required on one trailer axle. This table also shows the weight and cost of the assemblies per

TABLE 14. EFFECT OF LOAD ON TIRE MILEAGE [1]



[1] REPRODUCED FROM A CHART FURNISHED BY THE U.S. RUBBER CO.

TABLE 15. RIM SIZES FOR TRAILER TIRES

Tire Size	Rim Size [1]	Wgt. [2]	Cost [3]	Required for One Axle		
				No.	Wgt. [2]	Cost [3]
10.00 × 20 11.00 × 20	7.33V(9-10")× 20	73#	$8.58	4	292#	$34.32
10.00 × 22 11.00 × 22	7.33V(9-10")× 22	87*	$8.93	4	348#	$35.72

[1] From "U.S. Rubber Co. Technical Manual," page 69.
[2] Actual scale weight.
[3] Cost delivered to Bristol, Virginia via less-carload freight.

TABLE 16. COMPARISON OF WHEEL ASSEMBLIES [1]

TYPE OF WHEEL	TIRE SIZE	REQUIRED FOR ONE AXLE ASSEMBLY									
		TIRES		WHEELS		HUBS & DRUMS		RIMS		TOTAL	
		WGT.	COST	WGT.	COST	WGT.	COST	WGT.	COST	WGT.	COST
SPOKE	10.00 × 20	496#	219.96	370#	56.90	*[3]	*[3]	292#	34.32	1,158#	311.18
DISC	10.00 × 20	496#	219.96	420#	65.12	300#	59.06	*[4]	*[4]	1,216#	344.14
SPOKE	10.00 × 22	545#	233.40	396#	63.50	*[3]	*[3]	348#	35.72	1,289#	332.62
DISC.	10.00 × 22	545#	233.40	468#	70.00	300#	59.06	*[4]	*[4]	1,313#	362.42
SPOKE	11.00 × 20	591#	247.40	370#	56.90	*[3]	*[3]	292#	34.32	1,253#	338.62
DISC.	11.00 × 20	591#	247.40	420#	65.12	300#	59.06	*[4]	*[4]	1,311#	371.58
SPOKE	11.00 × 22	633#	320.80	396#	63.50	*[3]	*[3]	348#	35.72	1,377#	420.02
DISC	11.00 × 22	633#	320.80	468#	70.00	300#	59.06	*[4]	*[4]	1,401#	449.86

1 COMPILED FROM DATA IN TABLES 11, 12, 13, AND 15.

2 COSTS ARE DELIVERED TO BRISTOL, VIRGINIA VIA LESS-CARLOAD FREIGHT.

3 HUBS AND DRUMS ARE INCLUDED WITH SPOKE WHEELS.

4 RIMS ARE INCLUDED WITH DISC WHEELS.

thousand pounds of weight carrying capacity. The number of revolutions per mile of travel are shown for each size of tire and a mileage index has been calculated, based on 1.000 as the index for 10.00 x 20 tires. The index of tire mileage is inversely proportional to the revolutions per mile, assuming that the tires are loaded to their rated capacities.

The selection of the proper size tires depends on the load carried and the severity of the service to which the tires are subjected. The 10.00 x 20 size is used by a majority of the trailer users and has been found to work very well in most cases. This size has therefore been selected for the trailer for all normal service applications. In cases where a larger size tire is necessary, the selection can be made according to the information presented in Table 17.

TABLE 17. SPOKE WHEEL ASSEMBLIES INCLUDING TIRES [1]

TIRE SIZE	RATED CAP. ON 4 TIRES	WEIGHT	COST ($)	WGT. PER 1000 LBS. OF CAP.	COST PER 1,000 LBS. OF CAP.	REV. PER MILE	MILEAGE INDEX [2]
10.00 × 20	16,000 LBS.	1,158 #	311.18	72.4 #	$19.45	512.	1.000
10.00 × 22	17,100 LBS.	1,289 #	332.62	75.4 #	$19.45	486.	1.053
11.00 × 20	18,000 LBS.	1,253 #	338.62	69.6 #	$18.81	497.	1.030
11.00 × 22	19,000 LBS.	1,377 #	420.02	72.5 #	$22.11	473.	1.082

[1] COMPILED FROM DATA IN TABLE 13 AND 16.

[2] BASED ON REVOLUTIONS PER MILE AND USING 1.000 AS INDEX OF 10.00 × 20 TIRE.

SELECTION OF THE LANDING GEAR

The landing gear on the trailer is used when the tractor is disconnected from the trailer. The landing gear is first cranked down so that its wheels are resting on the ground and then the fifth wheel of the tractor is unlatched and the tractor driven out from under the trailer. This leaves the trailer resting on the tires mounted on the rear axle and on the small steel wheels of the landing gear.

A schematic drawing of a horizontal type landing gear is shown in Table 18A on page 37. When not in use, the legs of the landing gear are folded up into the position indicated by dotted lines on the drawing. The landing gear is operated by a hand crank located on the right (curb) side of the vehicle and is driven by a gear and screw type mechanism.

A similar drawing of a vertical type landing gear is shown by the drawing in Table 18B on page 37. When not in use, the wheels lift up into the position indicated by dotted lines. The landing gear is driven by a hand crank and has a pinion, self-locking worm, and rack mechanism.

The horizontal type landing gear weighs 290 pounds and costs $39.97 delivered to Bristol, Virginia via less-carload freight while the vertical type weighs 275

TABLE 18A. HORIZONTAL LANDING GEAR

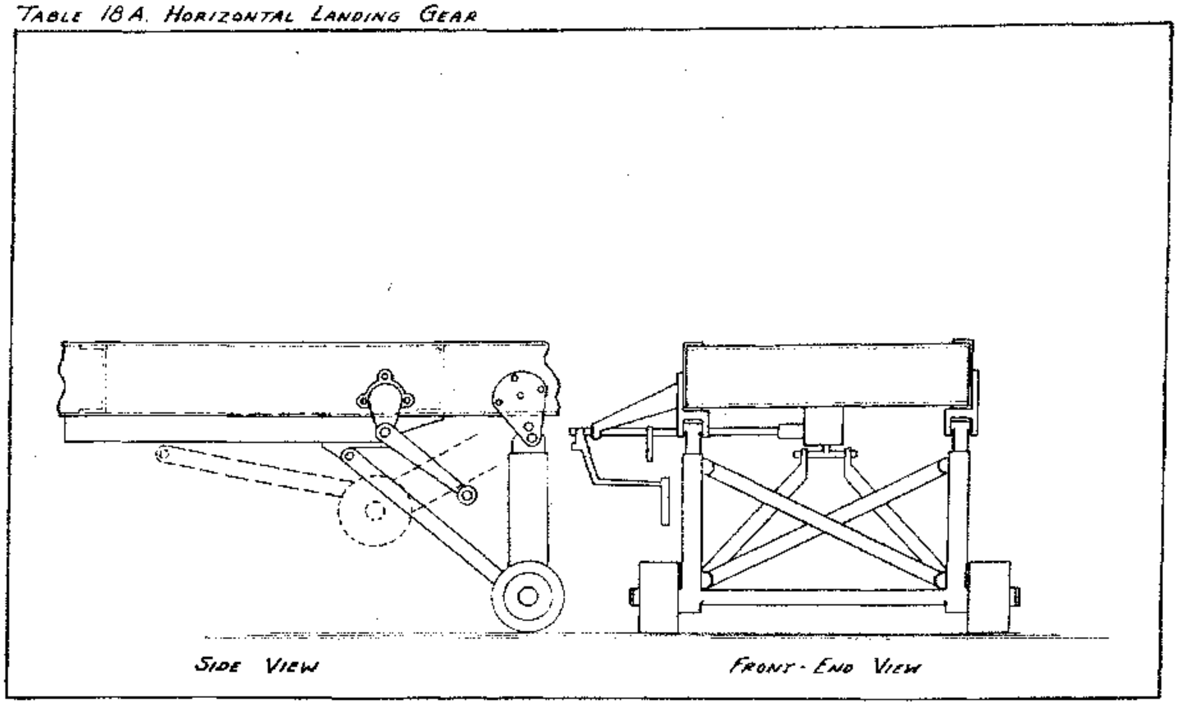
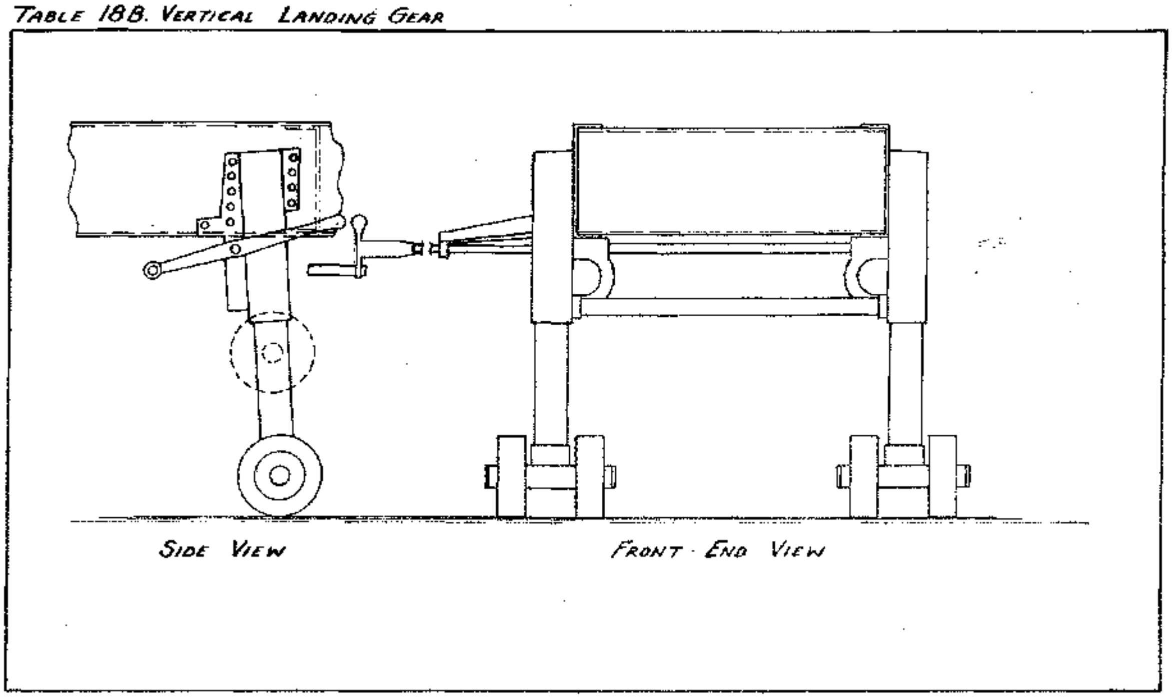


TABLE 18B. VERTICAL LANDING GEAR



pounds and costs $49.58 delivered. The vertical type has been selected for the trailer on account of its lesser weight, although it is more expensive. This extra cost is partially offset by the fact that the channels on which it is mounted need only a small cross channel for bracing while the horizontal type requires two heavy cross channels for mounting the screw mechanism. This is illustrated by the drawings in Tables 18A and 18B. This extra channel on the horizontal type landing gear further increases the weight difference between the two types and thus further helps justify the selection of the vertical type.

BRAKES

Trailers require power operated brakes on account of the very large weight of the trailer as well as the fact that power brakes are a part of the I.C.C. brake requirements. The two principal types of power brakes for trailers, vacuum and air, and the newer electric brakes will each be discussed briefly.

Vacuum trailer brakes take advantage of the fact that the engine of the tractor can be made to serve as a good vacuum pump by utilizing the vacuum available in the intake manifold. In this braking system air is pumped by the engine from out of one side of a flexible diaphragm and atmospheric pressure is applied on the other side of the diaphragm. This moves the diaphragm, thereby moving a rod connected to the diaphragm. This rod is connected to a lever which in turn is attached to the brake cam rod of a simple mechanical brake. The brake is thus applied when a vacuum is created on one side of the diaphragm. The maximum pressure on the diaphragm in this system is about thirteen pounds per square inch[1], so it is necessary to use a diaphragm of large area to obtain the required braking power.

1. See "Brakes", published by the American Trucking Association, page 39.

Air brakes operate in much the same manner as vacuum brakes except that the diaphragm is operated by compressed air supplied by an air compressor mounted on the tractor. A pressure of about sixty pounds per square inch is used on this system, so the area of the diaphragm can be considerably smaller than that of the diaphragm used in the vacuum system.

Electric brakes use power from the electrical system of the tractor to energize a magnet built into the wheel. When the magnet is energized it pulls a disc (rotating with the wheel) against the magnet. This drags the magnet along in the direction of rotation of the wheel and a lug attached to the end of the magnet presses against the brake cam, thus operating the cam and applying the brake. Electric brakes have been used on many small size trailers used by the armed forces of the United States but have not yet been used much on commercial trailers. Since they are not favored generally by trailer operators, they will not be considered further.

By referring to Table 10 on page 25 it will be seen that a 17,000 pounds capacity axle complete with air brakes and air brake kit costs $31.81 more than the same size axle equipped with vacuum brakes. Furthermore, it is necessary to pay about $200.00 to have a tractor equipped with air brakes and connections for trailer air brakes.

The weights of the axles equipped with air and vacuum brakes are practically the same, as is shown by Table 10.

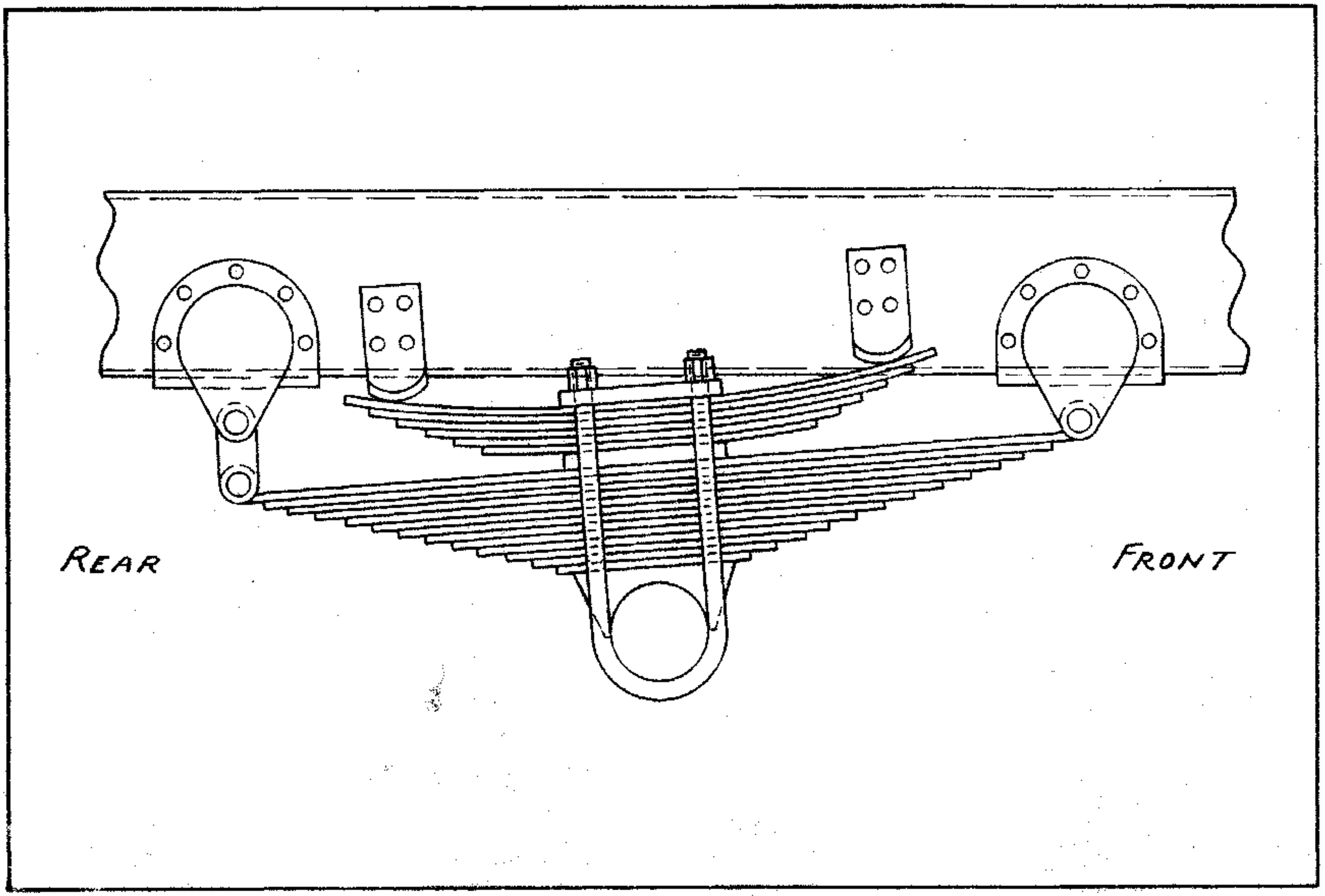
Air brakes are much easier to maintain than vacuum brakes. The compressor will furnish a large volume of air so that a small leak will not affect an air brake system much whereas the same size leak will be completely disastrous to the operation of the vacuum system. The vacuum system operates at a very low pressure and as a result, the check and synchronizing valves easily become clogged with any foreign material that may get into the system, causing the brakes to become inoperative. Thus it may be seen that the air brake system is much easier to maintain than the vacuum system.

Reliability in trailer brakes is of prime importance as safety of life and property depends on it. Air brakes are the most reliable and powerful brakes ever used on trailers. At the same time they will usually apply much more smoothly than vacuum brakes. These reasons will explain why trailer users are willing to pay the necessary extra cost to get air brakes and why approximately eighty per cent of the trailers built are now equipped with air brakes.

DESIGN OF THE SPRING ASSEMBLY

The Hotchkiss-drive type of spring assembly was one of the first used on trailers. An illustration of this type of spring assembly is shown by the drawing in Table 19 on page 43. The front end of the main spring is attached directly to the front spring hanger by means of a pin. The rear end of the spring is attached to the rear spring hanger through a connecting link (called a shackle) and this shackle compensates for the length of the spring varying as it deflects. The auxiliary spring acts against two angle-shaped brackets attached to the trailer frame. Alignment of the axle is secured by loosening the U bolts, slipping the axle to the proper place, and then retightening the U bolts. The towing stresses are carried from the front hanger through the spring to the axle. As the position of the axle depends solely on the friction between the bottom of the main spring and the top of the axle saddle (sometimes called spring seat), any looseness of the U bolts or heavy shocks due to hitting bumps will cause the axle to be knocked out of alignment. The towing stresses also often result in broken front spring eyes and sheared spring center-bolts. The many disadvantages of the Hotchkiss-drive spring assembly have led to its abandonment by most trailer manufacturers, even though it is the cheapest assembly of all the different types.

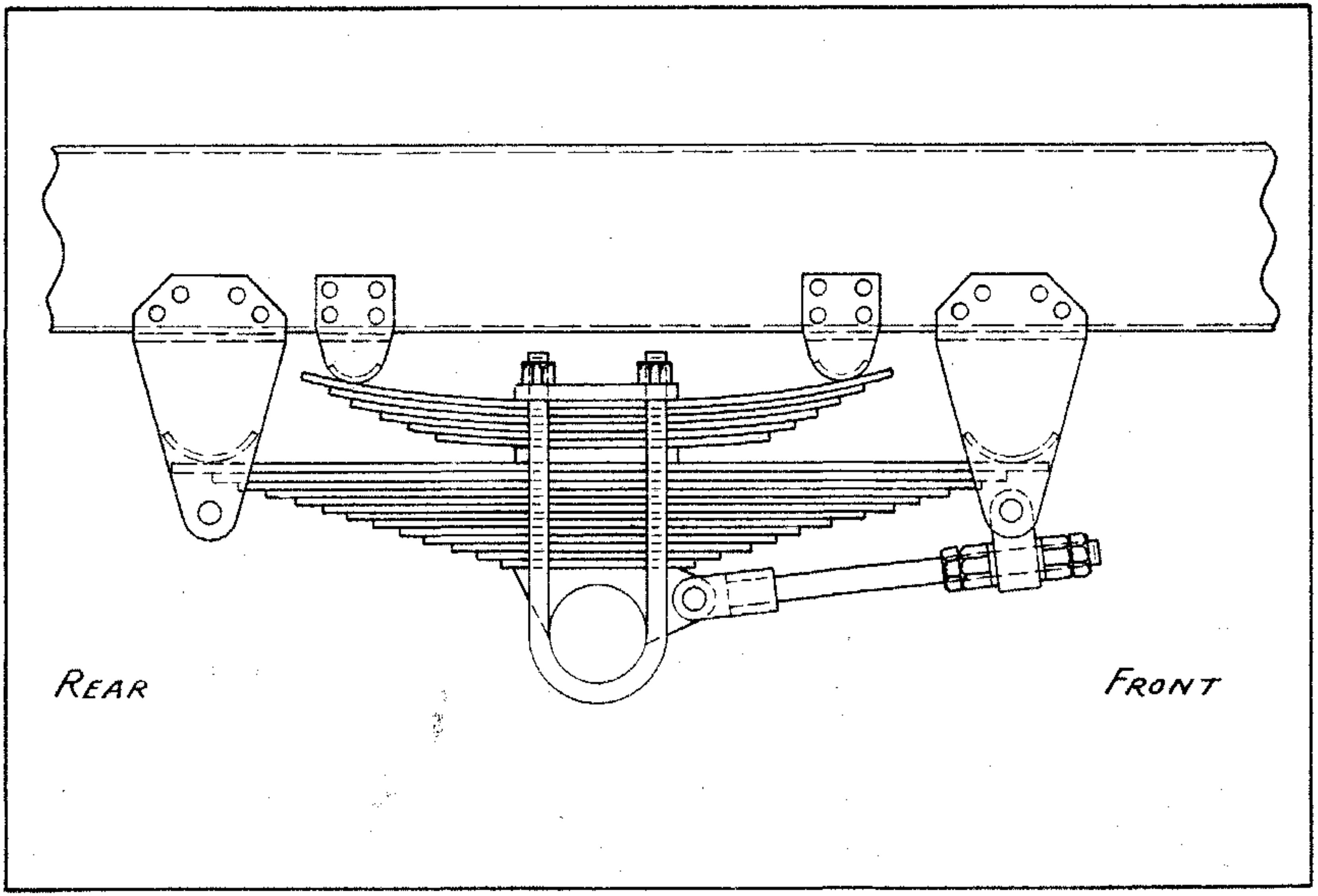
TABLE 19. HOTCHKISS - DRIVE SPRING ASSEMBLY.



The radius rod type of spring assembly was developed to provide a positive and adjustable means of securing axle alignment. A radius rod spring assembly is shown by the drawing in Table 20 on page 45. The radius rod should be located so that the towing stresses are carried directly from the front spring hanger through the radius rod to the axle saddle which is welded directly on the axle. In this way the radius rod carries the entire towing stress and the spring does nothing but cushion the load. The radius rod should be located so that it is as nearly horizontal as possible so that the small amount of axle movement caused by the radius rod swinging through an arc, as the spring deflects, will be as little as possible.

The main spring shown in Table 20 is of the slip-end type. While this type is less expensive than the shackled type, it is subject to excessive wear where the spring bears on the spring hanger. This is especially true in sections of the country where the trailer runs over dusty roads and the soil is of a sandy type. Dust gets between the spring and the hanger and acts as an abrasive, causing the top leaf of the spring to wear through and necessitating replacement of the spring. This condition is always true of the auxiliary springs on any type of spring assembly, but the wear is not nearly so severe as that on the main spring, since the auxiliary carries the smaller part of the load and the unit pressure of the auxiliary spring against the spring stop is much less than

TABLE 20. RADIUS ROD SPRING ASSEMBLY (SLIP-END SPRINGS).



that of the main spring against the spring hanger. It is not possible to use shackles on the ends of the auxiliary springs as on the main spring.

A radius rod spring assembly using shackled main springs is illustrated by the drawing in Table 21 on page 47. While this type of spring assembly is both heavier and more expensive than any other type, it is so much more satisfactory in operation and has found such great favor among trailer users that it has been selected as the assembly to use on the trailer. The various components of this spring assembly will now be taken up in detail.

The axle saddle has been designed so that it can be arc welded from sheared and flame-cut soft open hearth steel plates. The design has been made strictly on a basis of what is required to do the job, based on the writer's experience. A detail drawing of the axle saddle is shown in Table 22 on page 48. The bronze bushing shown in the drawing will also be used at the front end of the radius rod, in the spring eyes, and in the spring shackles. A bronze spacer washer will be used on each side of the axle saddle, between the saddle and the radius rod. The washer should be made from S.A.E. 64 bronze and should be made 3/16" thick x 1 3/16" inside diameter x 2" outside diameter. This bronze washer will also be used as a spacer at the front end of the radius rod, at the spring eyes, and at the shackles.

TABLE 21. RADIUS ROD SPRING ASSEMBLY (SHACKLED SPRINGS).

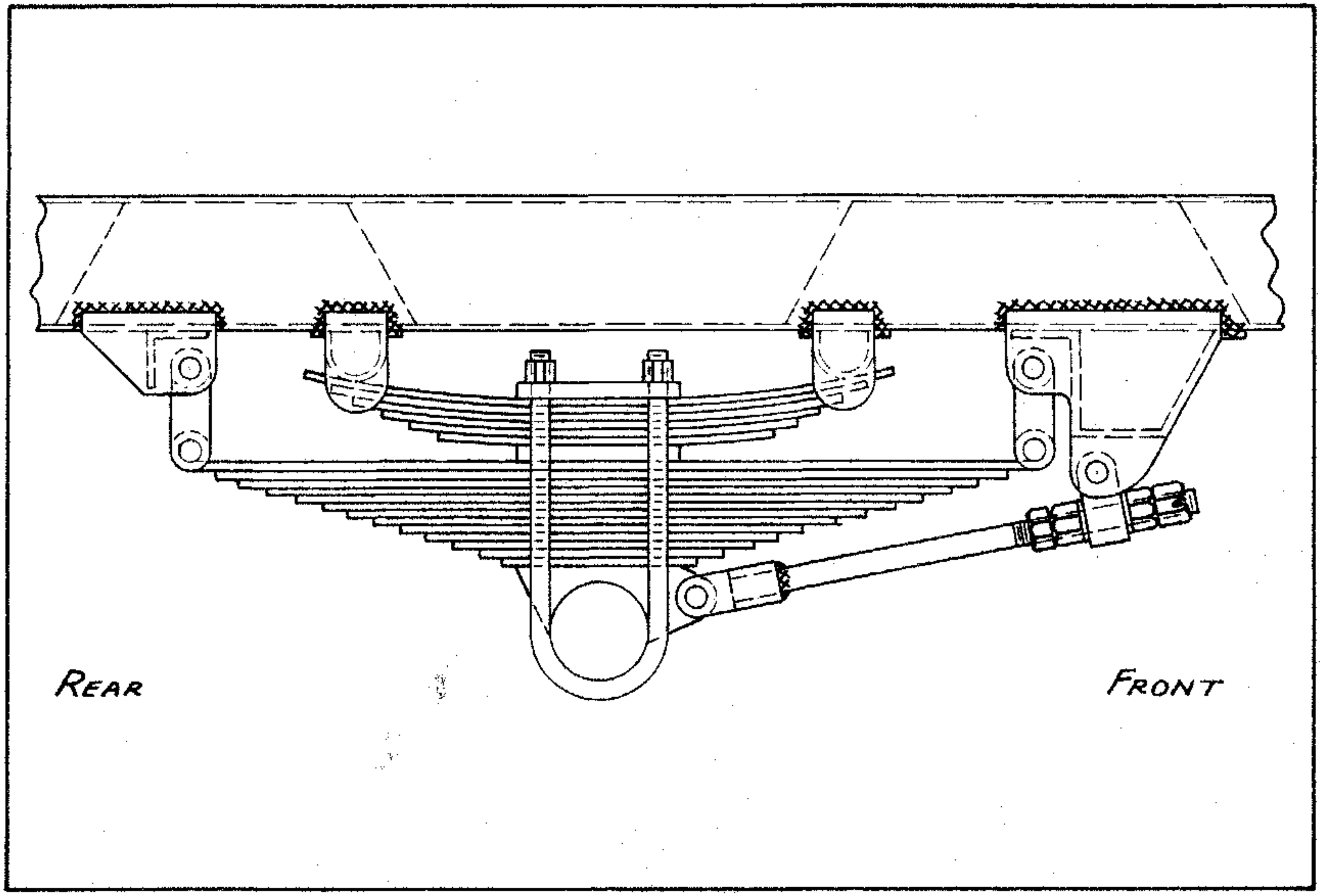
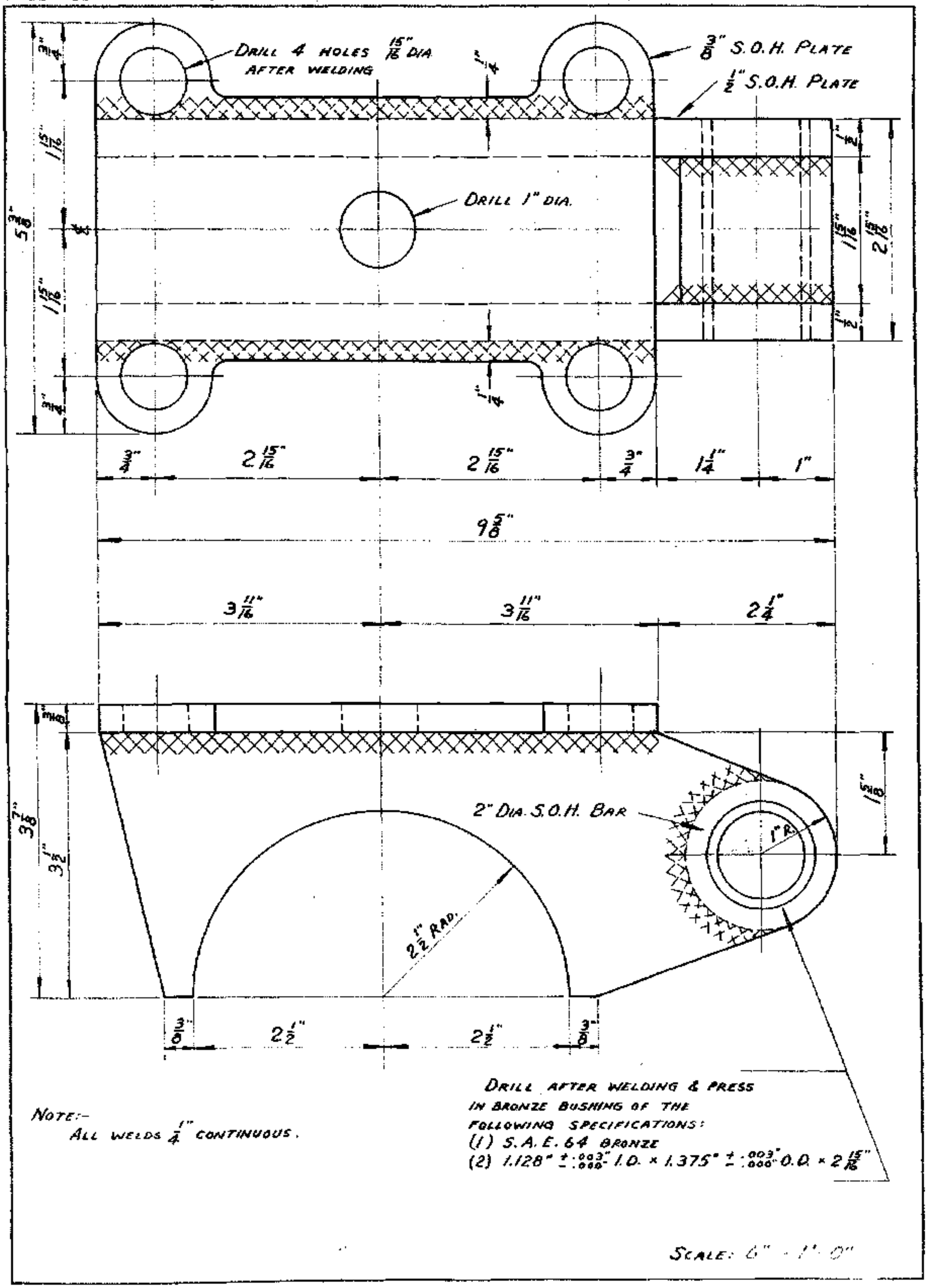


TABLE 22. AXLE SADDLE.



The cast steel radius rod is shown by the drawing in Table 23 on page 50. This design is a result of a process of improvement in past designs made necessary by ever-increasing loads. Particular care has been taken to avoid sudden changes in section and in direction of stress in order to prevent fatigue failures which might be caused by such stress-raisers. The writer intends to study this particular part in the near future by means of the brittle lacquer method of stress analysis (Stresscoat method) with the purpose of further improving the design.

The cast steel front radius rod connector is shown by the drawing in Table 24 on page 51. This part can not be varied much as it has very definite space limitations into which it must fit. The part shown has been very satisfactory in service. The grease fitting is necessary because there is a twisting of the radius rod in the front connector when the spring on one side of the trailer is deflected more than the spring on the other side, as is the case when a curve is rounded, even at moderate speed.

Table 21 on page 47 will show that a washer is used on the radius rod on both sides of the front connector. This is a steel washer 1/4" thick x 1 9/16" inside diameter x 2 1/2" outside diameter. The front connector is located by two 1 1/2" American standard light hex nuts as is shown by the drawing in Table 21. It is locked in place by an 1 1/2" American standard light jam nut on the rear side and

TABLE 23. CAST STEEL RADIUS ROD.

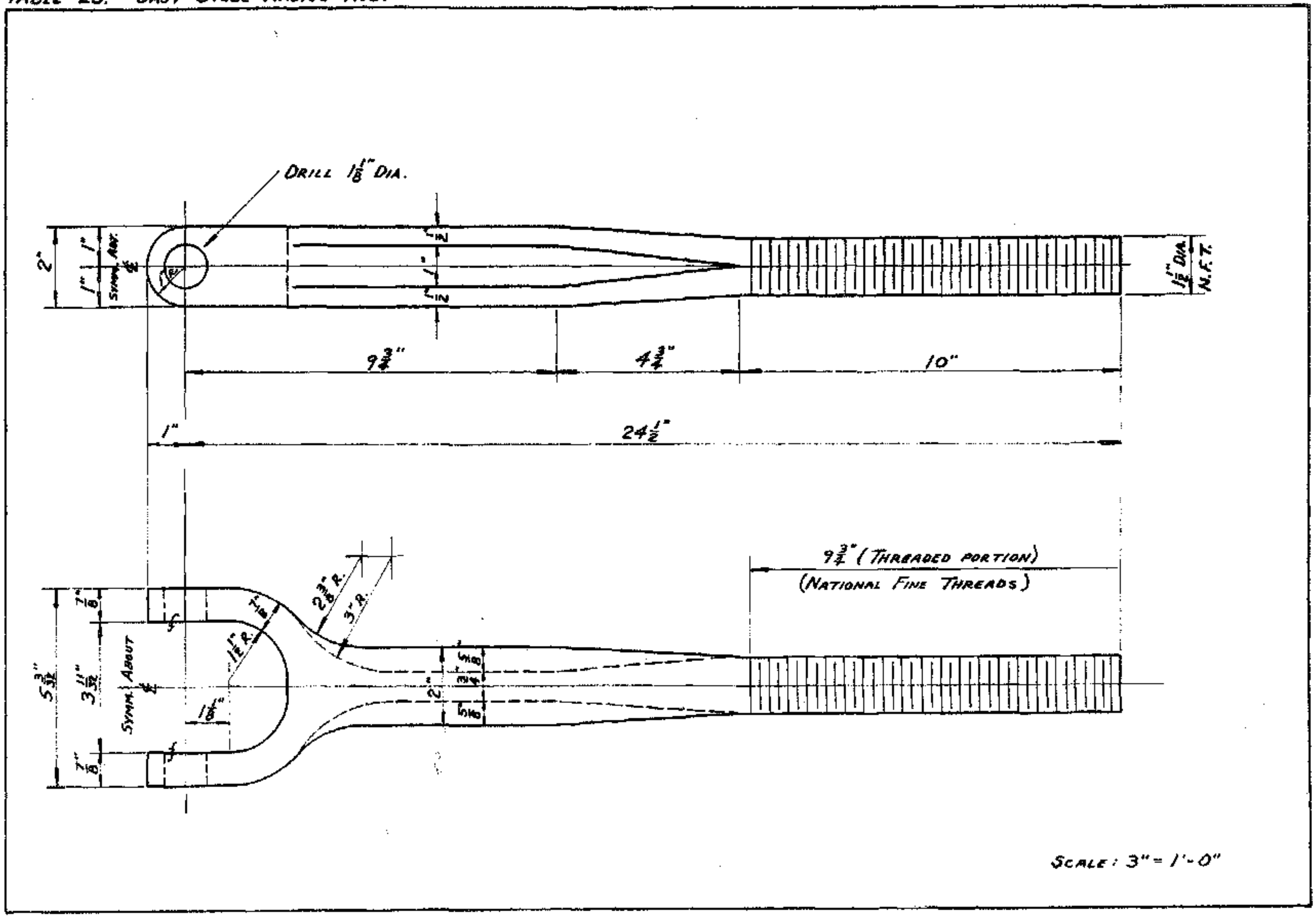
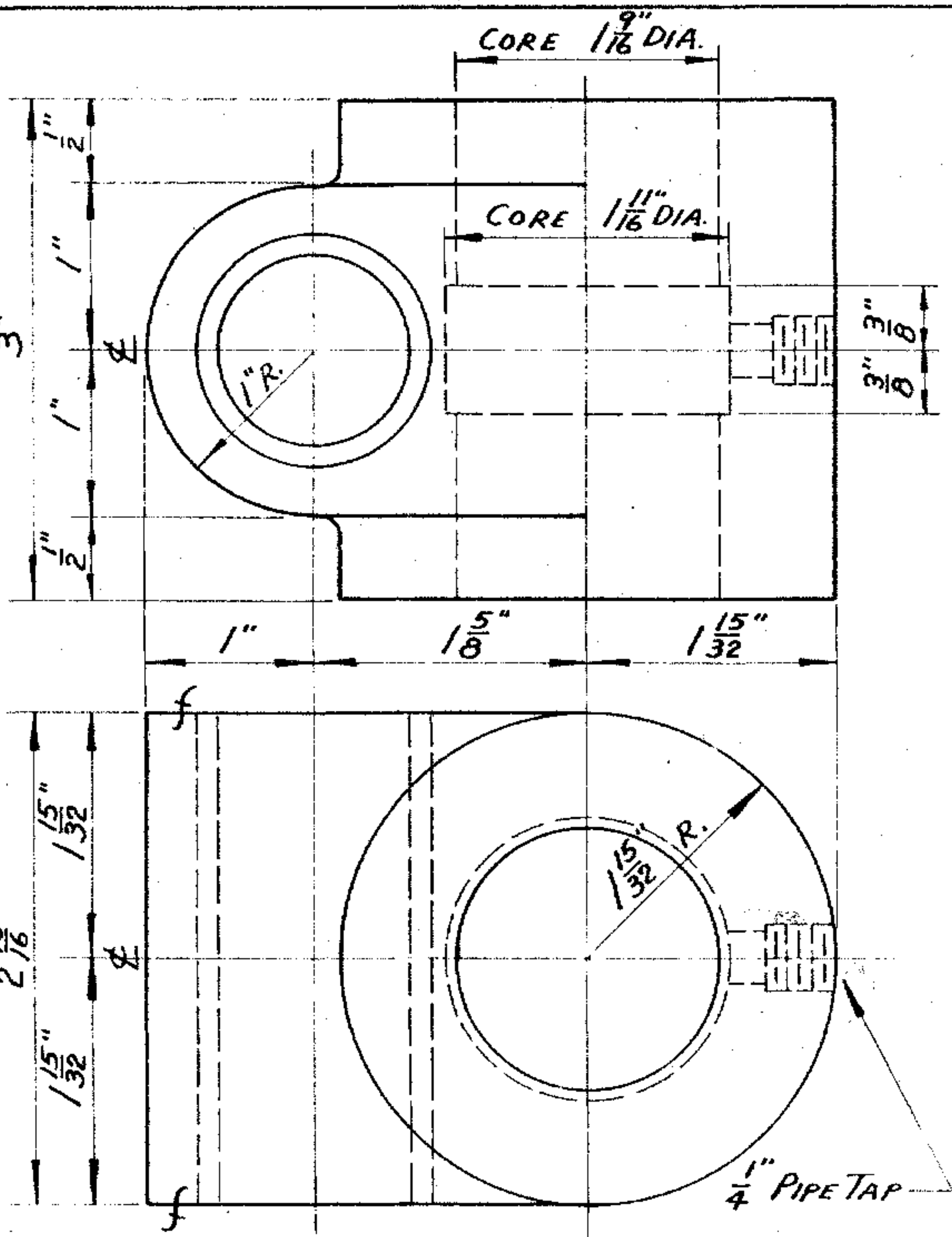


TABLE 24. CAST STEEL RADIUS ROD CONNECTOR.



SCALE: FULL SIZE

NOTE:-

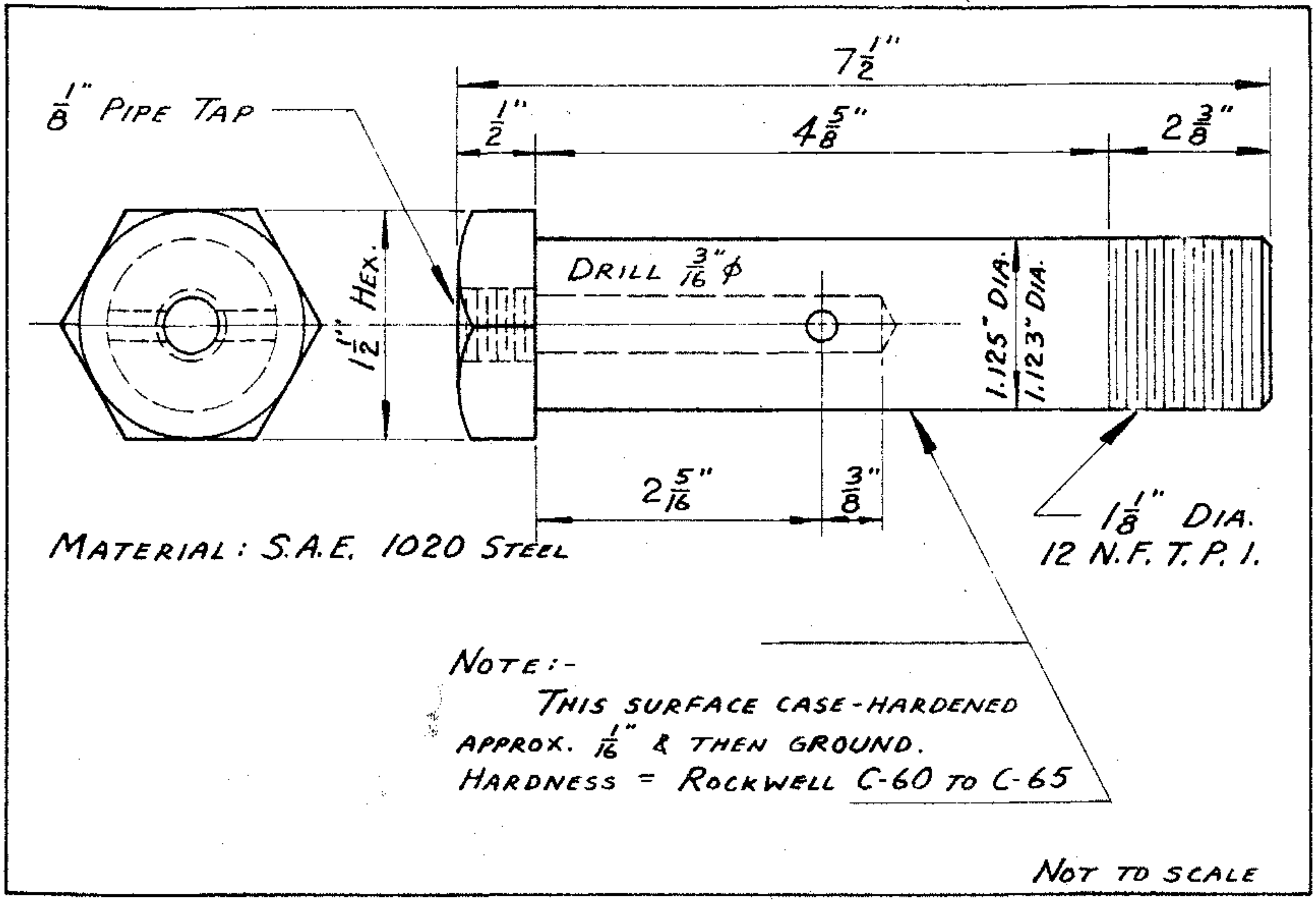
DRILL & PRESS IN BRONZE BUSHING SAME AS IN AXLE SADDLE SHOWN IN TABLE 22.

by an 1 1/2" American standard light slotted nut on the front side. The nuts are adjusted to secure exact alignment of the axle and then a hole is drilled through the radius rod so that a cotter can be used to secure the slotted nut in place.

The radius rod is held in place by the shackle pin shown in the drawing in Table 25 on page 53. This pin has been used many years and the design has not been changed. It is held on by an 1 1/8" Elastic stop nut. This is a special self-locking nut manufactured by the Elastic Stop Nut Corporation and uses a piece of fiber crimped into the top of the nut as the locking means.

Since the springs are standard automotive components manufactured by companies specializing in this work, only a summary of the specifications of the springs will be given. The main spring has 13 leaves, 3/8" x 3", and is 45" long between the centers of the eyes when loaded. The load carried by the main spring is 5800 pounds and the flexibility is 1895 pounds per inch of deflection (average). The eyes of the spring are equipped with the same bronze bushings used in the radius rod connectors, the axle saddle, and the shackles. The auxiliary spring has 6 leaves, 5/16" x 3", and is 34" long when loaded. It carries 1900 pounds of the load and has a flexibility of 1900 pounds per inch of deflection at an effective length of 29". Both the main and auxiliary springs are bolted together with 1/2"

TABLE 25. SHACKLE PIN.



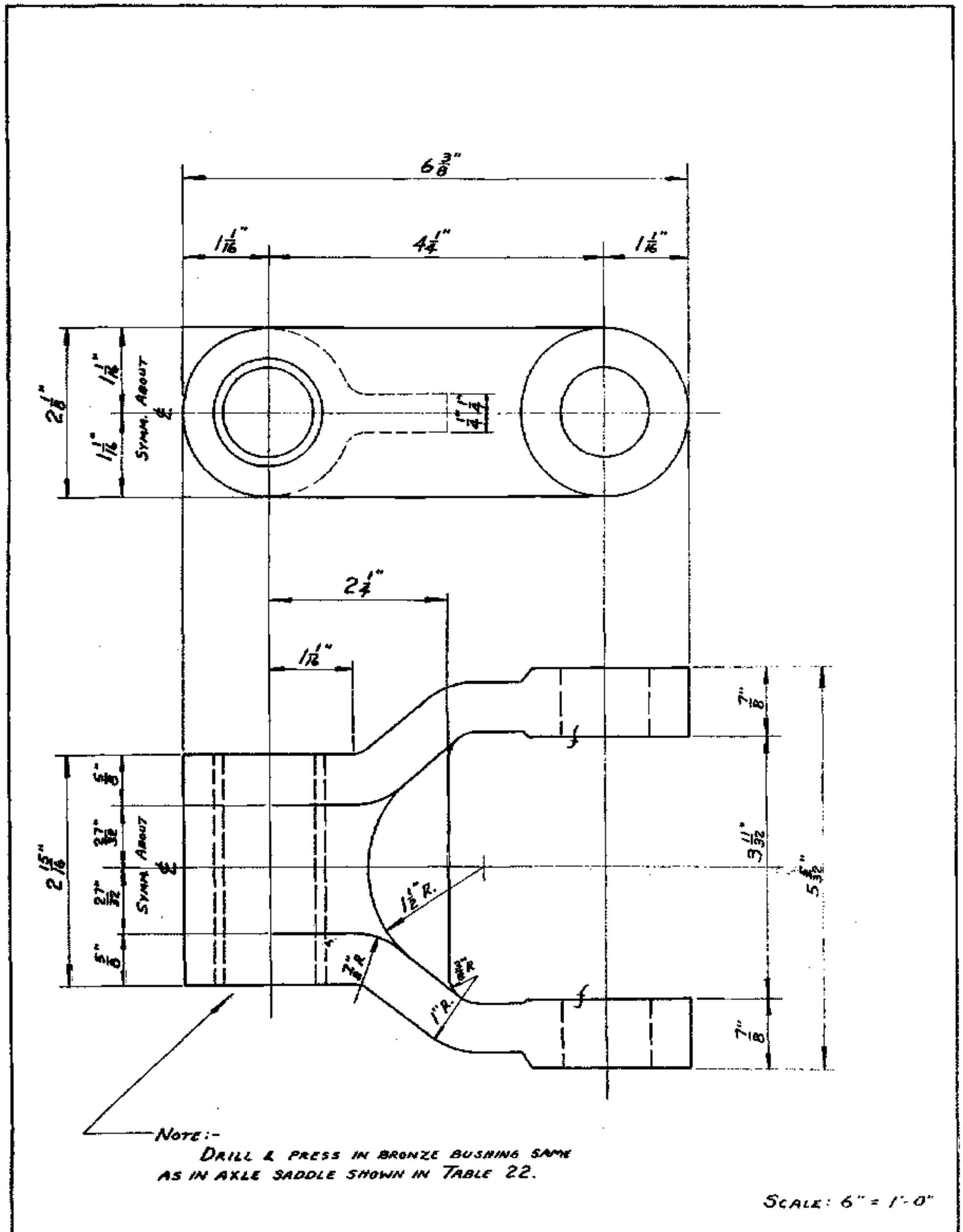
diameter center bolts. The weight of two complete springs is 416 pounds and the rated capacity is 15,400 pounds at the normal deflection of three inches. With a deflection of only 1/4" beyond normal the load rating increases to 17,300 pounds so the spring should be strong enough for all normal size loads. The spring will stand a total of five inches of deflection without evidence of permanent set.

The cast steel spring shackle is shown by the drawing in Table 26 on page 55. The design of this shackle is the result of many changes as loads on trailers increased. As in the case of the radius rod, care was taken to avoid sudden changes in section and in the direction of stress to give maximum resistance to fatigue failures. The writer also plans to make a study of the shackle by the Stresscoat method in the near future, although the design shown has proven to be very satisfactory.

The spring pressure plate is shown by the drawing in Table 27 on page 57. It is simply a piece on soft open hearth plate with holes drilled for the U bolts and rounded on the corners for appearance. The 9/16" hole slightly off the center is for the nut on the auxiliary spring center bolt. The 3/4" thickness plate has been found to be the thinnest that can be used without taking a permanent set.

The U bolts are made from S.A.E. 3135 (or N.E. 8630) steel bars. They are threaded and then bent to a

TABLE 26. CAST STEEL SHACKLE.



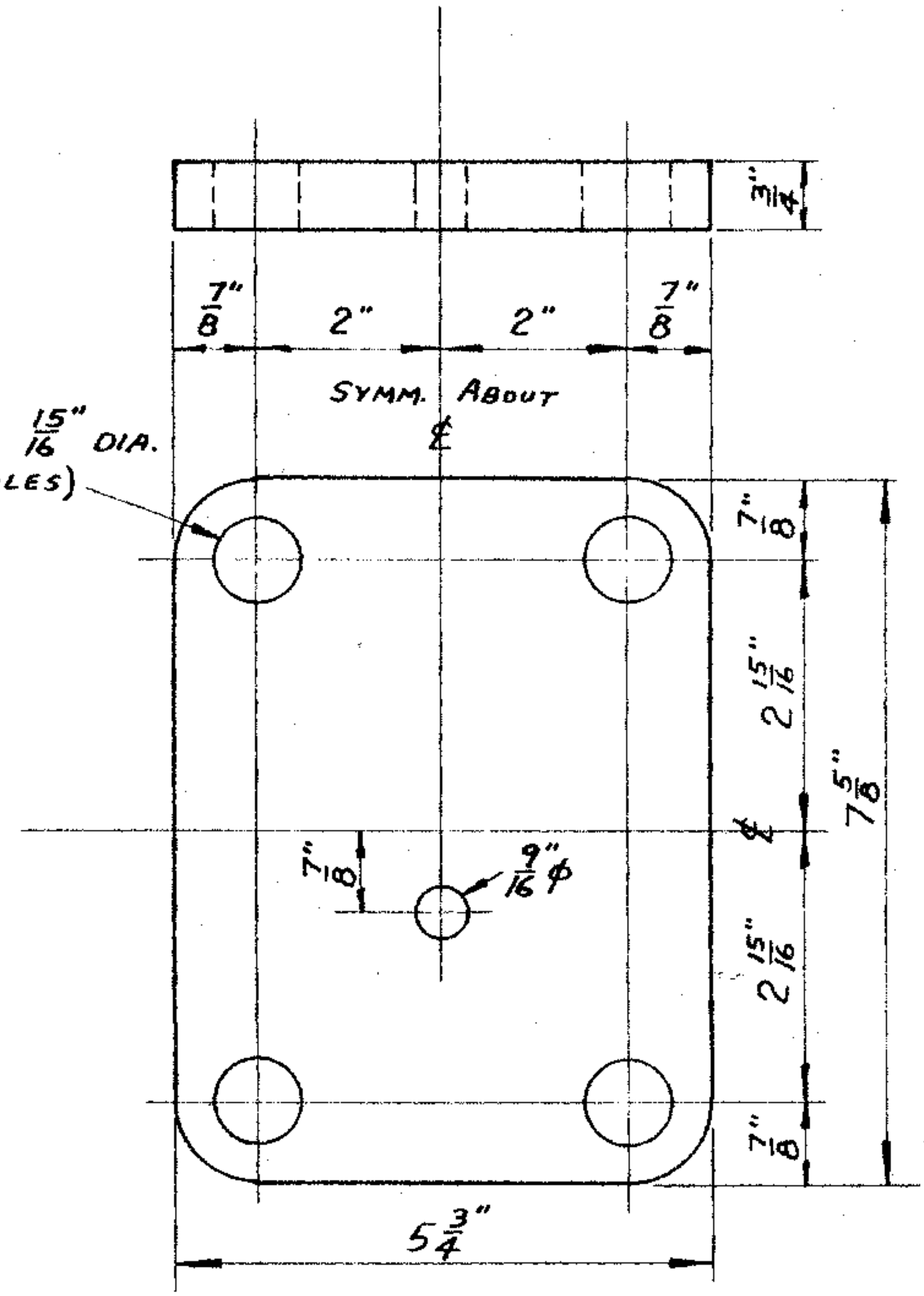
5" inside diameter to fit around the axle. This is shown in the drawing in Table 21 on page 47. The U bolt nuts are S.A.E. standard high hex nuts with National fine threads.

The grease fittings used in the shackle pins and in the front radius rod connector are Alemite no.1980 straight fittings with 1/8" pipe threads.

The weight and cost of the spring assembly will be included in the weight and cost estimate of the completed trailer.

The design of the spring hangers will be worked out along with the structural members of the trailer, as the design of one will affect the design of the other.

TABLE 27. SPRING PRESSURE PLATE.



MATERIAL: S.O.H. PLATE

SCALE: 6" = 1'-0"

TYPES OF TRAILERS

Highway freight trailers consist of two principal kinds, the frame type and the frameless type. There are thousands of each type on the road and each will be discussed, although the newer frameless trailer is gradually supplanting the old frame style.

The frame style of trailer consists mainly of two very heavy channel-shaped longitudinal members having an offset near the front of the trailer. In this type of trailer the fifth wheel plate and the spring hangers attach directly to the main frame members, which are spaced transversely by cross zees or channels. Heavy cross members extend transversely to the sides of the body and these cross members are used to support both the body and the floor. Most frame style trailers have excessive deflection in the frame due to the extreme length of the trailer and small depth of section and this continual deflection of the frame results in considerable working of the parts in the sides and roof of the body, thus shortening the life of the body. The main advantage of a frame type of trailer is that it will withstand wrecks which will tear off a complete side of the trailer while a frameless type trailer will fold up completely if one of the sides is torn off and sometimes if only the roof is torn off. The main disadvantage of the frame type is its heavier weight. A thousand pounds more weight in a trailer means that a thousand pounds less cargo can be hauled on each trip the trailer makes during its life. When this is figured at the lowest freight rate, it totals so much that practically every trailer user is very eager to have his trailers as light as possible and still give good service.

The frameless type trailer utilizes the sides of the trailer to serve as beams to carry the load. In the frameless type trailer, members run transversely from one side to the other and carry the floor as well as the structure on which the spring assembly is mounted and also the structures on which the landing gear and fifth wheel plate are mounted. Since the side of the trailer is a beam about eight feet high, it is very stiff elastically and does not have enough deflection to be seen with the eye, even under full load. Since the frameless trailers have always proven to be lighter than the frame type, the trailer under consideration will be designed as a frameless one and the design of all the component parts will be taken up in detail.

DESIGN OF STRUCTURAL ELEMENTS OF THE FRAMELESS TRAILER

The normal load on a highway freight trailer will vary from 20,000 to 30,000 pounds, depending on the nature of the cargo hauled and on how carefully the user checks the weights of the various items on his manifest of the cargo. Some trailer users check weights very carefully, some only estimate the weight of the cargo by observing spring deflection, while others simply do not pay any attention to how much weight they put into the vehicle. Obviously, the trailer must be designed with adequate factors of safety to take care of all the abuse to which it will be subjected. For the purposes of design a normal payload will be taken as 30,000 pounds.

While the trailer must be designed to take care of overloads, it is not necessary to increase structural weight too much to do this. In the case of using stressed-skin side construction, the trailer will be designed so that the sides will not buckle under normal loads. In the case the sides are overloaded, they will buckle from shear strains and will show wrinkles. This can not be tolerated under normal load conditions for the sake of appearance, but in case the trailer is badly overloaded, the operator is usually aware of this and will not complain if such wrinkles appear under these conditions. It might be mentioned that such wrinkles show up on the wing surfaces of commercial air-line planes while in flight. These cause no concern to the average passenger but certainly do not add to the comfort of an engineer who may be riding the plane. As the effect of overloads is different on the various structural members of the trailer, it will be taken into consideration in one way or another when designing each of such members.

Concentrated loads are not much of a problem when designing freight trailers. The cargo is nearly always of such nature that it is spread pretty well over the length of the trailer, even when machinery is being hauled in the trailer. Trailer owners are becoming increasingly aware that the life of the trailer can be greatly prolonged if cargo of a concentrated nature is placed on long timbers to distribute the load properly. The writer recently found a case where heavy guns were breaking up the trailer floors. The problem was solved by simply placing some timbers at the proper places to distribute the load.

A trailer running along a highway receives a varying series of impacts. On a concrete highway where the main source of impacts is from the expansion joints, the trailer will receive up to about 300 impacts per mile of travel. The number of impacts received on a macadam road will depend entirely on the condition of the road, but the number of impacts will probably be more numerous and severe than those received on the concrete highway. If a trailer is run only 200 miles a day for 5 days a week and for 50 weeks a year, it will travel 50,000 miles in a year. This is about the average travel for a freight trailer, although some will travel nearly twice that much. If the trailer receives only 200 impacts per mile for 50,000 miles, then it has received 10,000,000 impacts in a year. This must be taken into consideration when designing all the structural members subject to fatigue failures; and the endurance limit, rather than the yield point, must be used in calculating the strength of the members.

Since the various structural members of the trailer are subject to indeterminate dynamic and impact loads, it is not possible to set up any definite factor of safety and use it in designing all members. Instead, it will be necessary to consider each member separately and to design each with a factor of safety commensurate with the actual experience of the writer as to what is required to work successfully in the particular application.

The properties of the various floor materials have been given in Table 5 on page 15 and white oak has been chosen as the floor material for the trailer for the reasons already stated.

The white oak for the floor lumber should be kiln-dried to about 12 % moisture content. It will then weigh approximately 3.644 pounds per board foot and will cost about $200.00 per thousand board feet when bought as finished and dimensioned lumber.

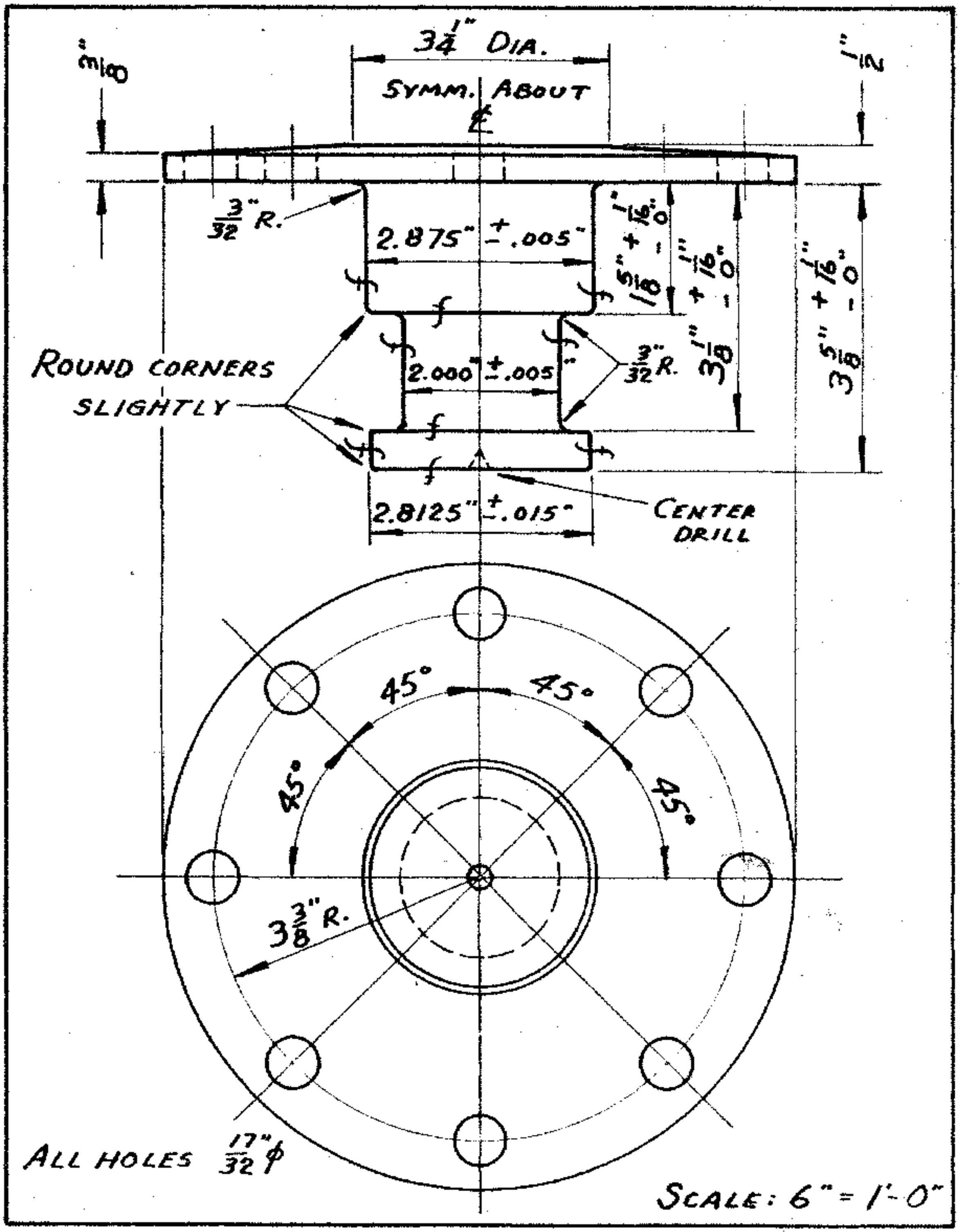
The floor should be made up in shiplap stock so that each board overlaps the adjacent board by about a quarter of an inch to prevent moisture from being thrown up through cracks between the floor boards. If the shiplap tongue is wider than a quarter of an inch it will get broken off too easily. The floor should be attached to the supporting cross members by bolts or self-tapping screws having flat countersunk heads. A Phillips recessed head should be used as it improves the appearance of the floor and facilitates the use of air-driven screwdrivers for tightening the bolts or screws. In either case, it is easier to tighten the fastening bolt or screw by driving the head from the inside of the trailer.

Before assembling the floor boards they should be dipped and allowed to stand for at least five minutes in a waterproof wood sealer. They should then be removed from the dipping tank and allowed to drain and dry. This sealing treatment prevents rot and preserves the lumber appreciably at very little cost. It also fills the pores of the lumber so that less primer and paint are required and the usual suction holes encountered when paint dries on unsealed wood are almost entirely absent. The sealer without any additional coating provides a nice finish for the part of the floor showing on the inside of the trailer as it gives the wood a light, pleasing amber color.

The kingpin transmits all the towing stresses from the tractor to the trailer and failure of the kingpin would cause the trailer to come unhitched from the tractor. For this reason the kingpin should be of high quality steel and a forging is much to be preferred for this purpose. A detail drawing of the kingpin is shown in Table 28 on page 65. The material is specified as S.A.E. 3140, since this steel has worked very well in service. The kingpin is machined to the standard dimensions adopted by the Society of Automotive Engineers because a pin of these dimensions will work on practically all lower fifth wheels manufactured in this country. Holes are provided so that the kingpin can be riveted to the upper fifth wheel plate, using eight 1/2" diameter rivets. It is desired to rivet the kingpin to the upper fifth wheel plate rather than weld it as the kingpin must be replaced every year or two due to wear and it is much easier to remove and replace if riveted on rather than welded.

The upper fifth wheel plate is usually made of 3/8" thick soft open hearth plate or else of 1/4" high tensile plate, with bracing to prevent excessive deflection. Both of these alternate designs have been proven by long service and represent the minimum sizes which can be used. Since the 1/4" low alloy high tensile steel plate with extra bracing is the lighter of the two, it will be used in the trailer. The dimensions of the fifth wheel plate will be shown in the drawing of the assembly of the upper fifth wheel section.

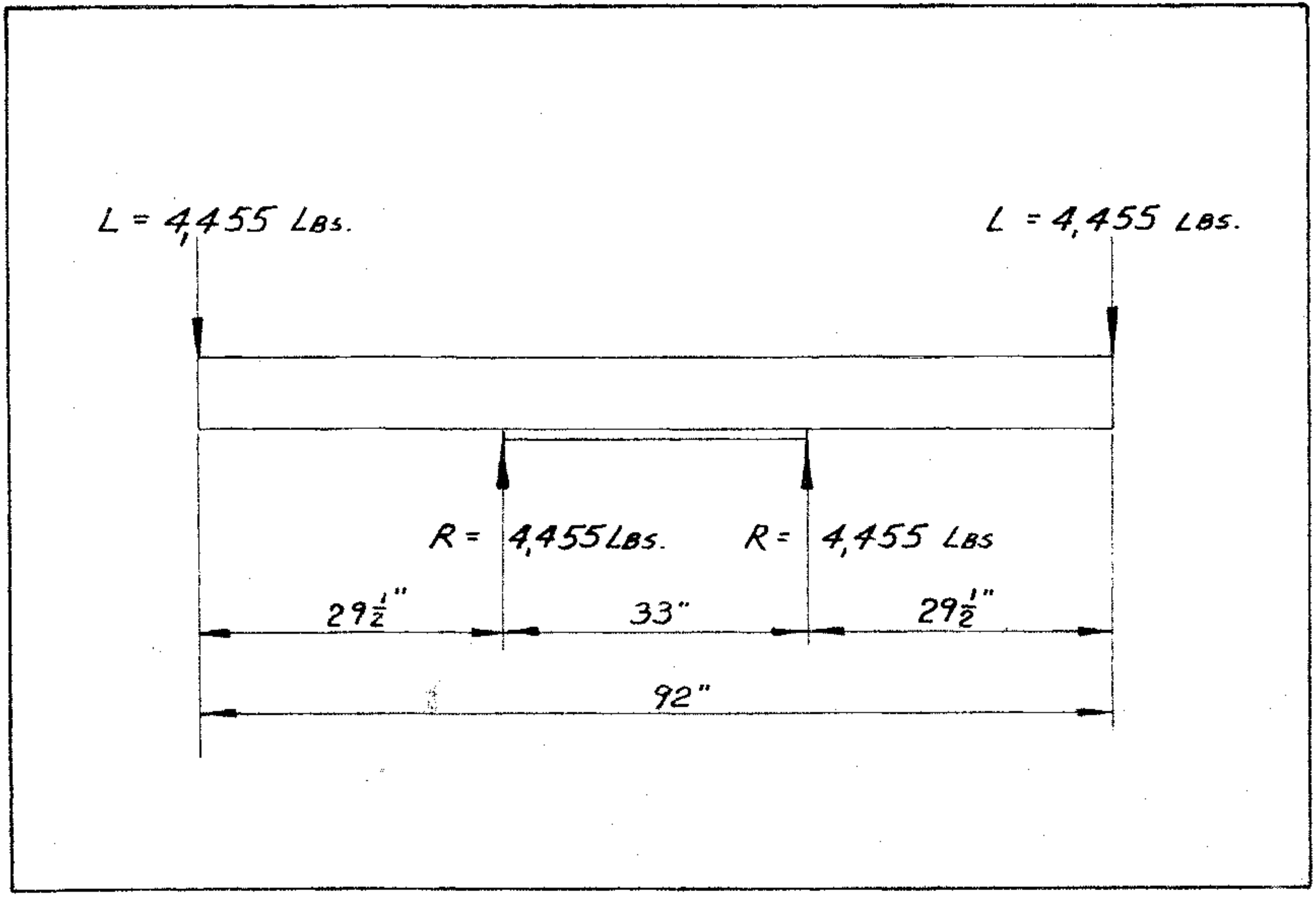
TABLE 28. FORGED KINGPIN[1] (S.A.E. 3140 STEEL).



[1] SEE "S.A.E. HANDBOOK, 1940 EDITION", PAGE 161, 1-40.

The upper fifth wheel plate is supported entirely by two heavy cross members running transversely from one side of the trailer to the other. Table 8 on page 21 shows that the maximum part of the payload carried at the fifth wheel plate is 15,320 pounds, with the payload assumed as being 30,000 pounds. This 15,320 pounds plus 2,500 pounds as the estimated weight of the trailer alone on the fifth wheel plate adds up to a total of 17,820 pounds load on the upper fifth wheel plate. Since this load is transmitted from the sides to the fifth wheel plate by two beams, then the force at each end of each beam is one-fourth of 17,820 pounds, which is 4,455 pounds. Table 29 on page 67 shows the load diagram of the beams. The dimension between the two reactions is taken as 33" because most lower fifth wheels are that wide, although some are slightly wider. The over-all dimension of 92" allows 2" on each side of the trailer for sides and rubrail, without exceeding the maximum over-all legal limit of 96". The beams will be made from low alloy high tensile steel. Reference to Table 4 on page 13 shows that such steels have an endurance limit of around 45,000 pounds per square inch. Since the two members under consideration are subject to both dynamic and impact loads, they are subject to fatigue failures and the endurance limit must be used as the maximum strength. Using a factor of safety of two and an endurance limit of 45,000 pounds per square inch, then the allowable unit stress is 22,500 pounds per square inch

TABLE 29. LOAD DIAGRAM ON FIFTH WHEEL PLATE SUPPORTING MEMBER.



for this particular application. The required size of the members is found by the following calculations:

Let M = maximum bending moment
S = required sectional modulus
p = maximum allowable unit stress = 22,500 psi.

$$M = (4455)(29.5) = 131,425 \text{ inch lbs.}$$

$$S = \frac{M}{p}$$

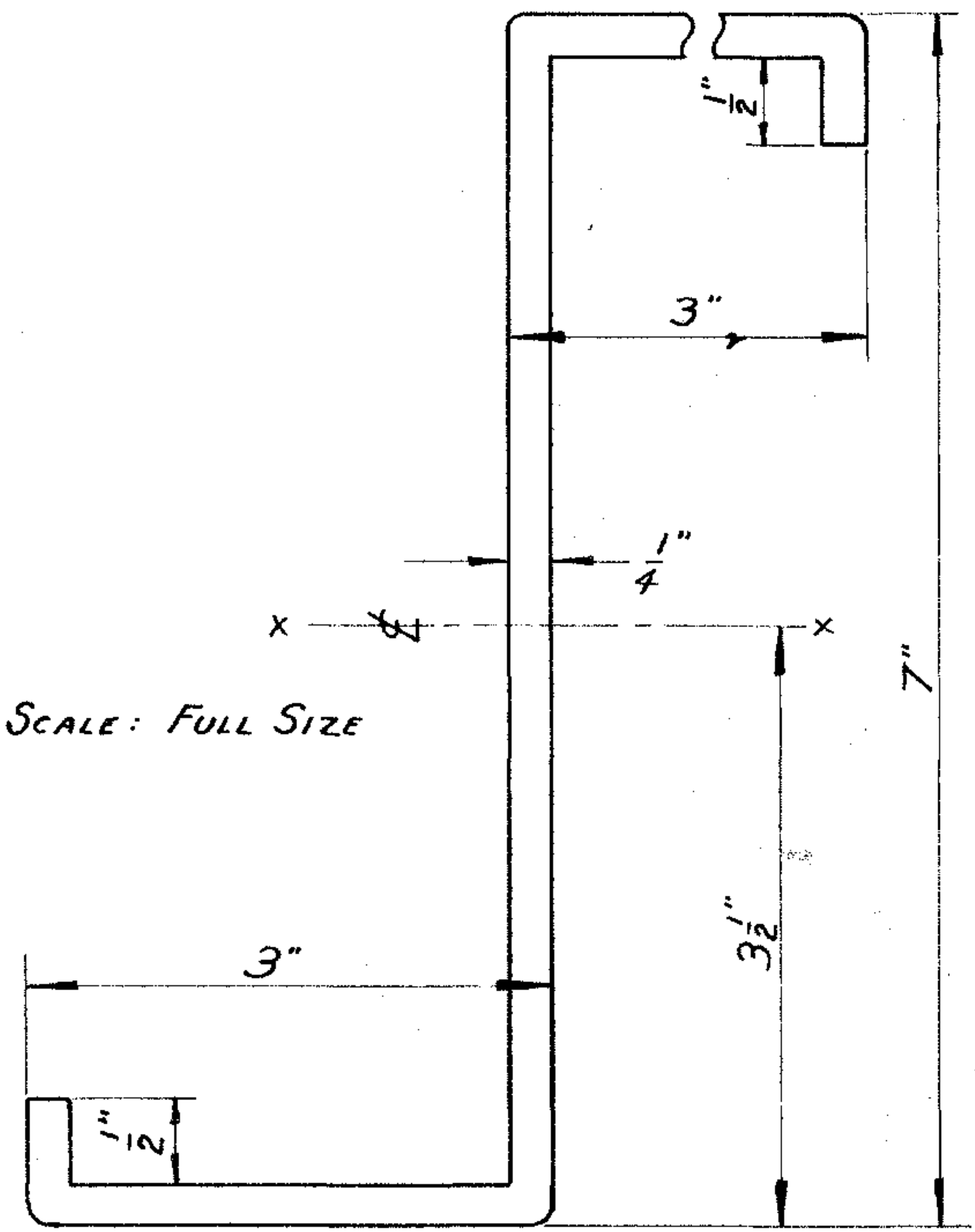
$$S = \frac{131,425}{22,500}$$

$$S = 5.841$$

A zee shaped member will be used and this is shown by the drawing in Table 30 on page 69. The legs of the zee are both flanged 1/2" more than the simple zee shape. This has been done to prevent the outermost fibers from having a sheared edge which might have some burrs which would act as stress-raisers and contribute to fatigue failures. The writer has seen this happen on a number of members and has found that the extra flange shown will usually cure such troubles. These small flanges will be neglected in calculating the sectional modulus of the member. The sectional modulus is found as follows:

Let I = moment of inertia about axis x-x
S = sectional modulus about the same axis
b = thickness of web in inches
d = depth of web in inches
a = area of flange
h = displacement of neutral axis in inches
c = distance from neutral axis to the most remote fiber

TABLE 30. FIFTH WHEEL PLATE SUPPORTING MEMBER.



MTL. – LOW ALLOY HIGH TENSILE STEEL, 1"/4 × 91 3"/4 LONG

$$I = \frac{b\,d^3}{12} + I_o + a\,h^2$$

Neglect I_o since it is very small

$$I = \frac{(.25)\ (7)^3}{12} + 2\ (.25)\ (2.75)\ (3.375)^2$$

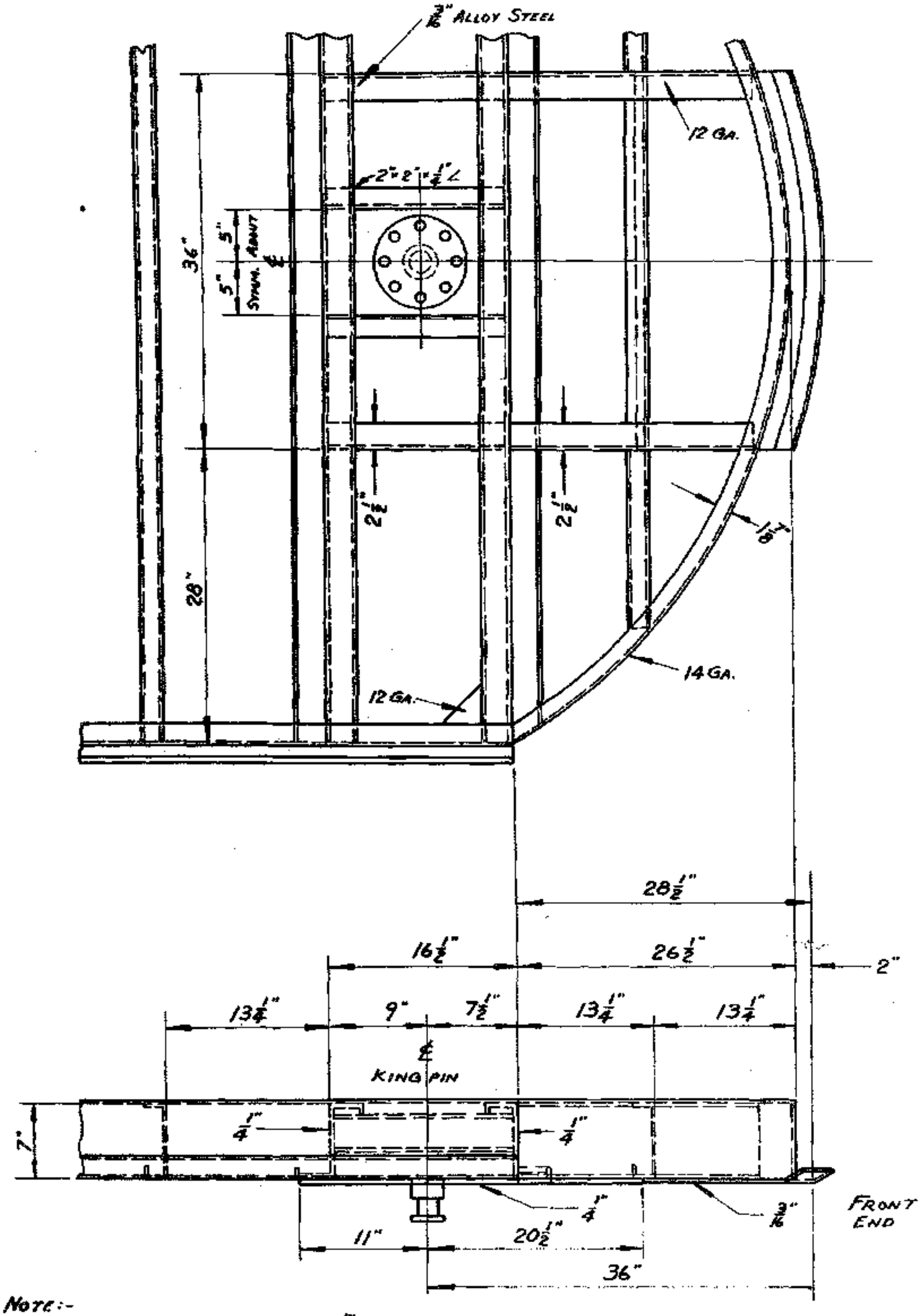
$$I = 22.84$$

$$S = \frac{I}{c} = \frac{22.84}{3.5}$$

$$S = 6.53$$

This is slightly more sectional modulus than the required 5.841 already calculated. In view of the fact that the factor of safety of two is as low as can be safely used on this particular member, the value calculated above seems very satisfactory and 1/4" thick material will therefore be used in these members. The assembly of the upper fifth wheel section of the trailer is shown by the drawing in Table 31 on page 71. The method of bracing the fifth wheel plate with 2" x 2" x 1/4" angles is shown. This method has been tried and found to be very satisfactory. The front part of the fifth wheel plate has been made from 3/16" thick material, as this is the thinnest that can be used here without taking a permanent set, as has been found from trial on actual vehicles. The other members which are not stressed extremely are made from the thinnest material possible, based entirely on the writer's experience. It should be mentioned that the front fifth wheel plate

TABLE 31. ASSEMBLY OF UPPER FIFTH WHEEL SECTION.



NOTE:-

ALL MATERIAL EXCEPT 2"×2"×1/4" ∠'S & KINGPIN IS LOW ALLOY HIGH TENSILE STEEL.
WELD TOGETHER WITH CONT. WELDS EQUAL TO THICKNESS OF MATERIAL.

SCALE: 3/4" = 1'-0"

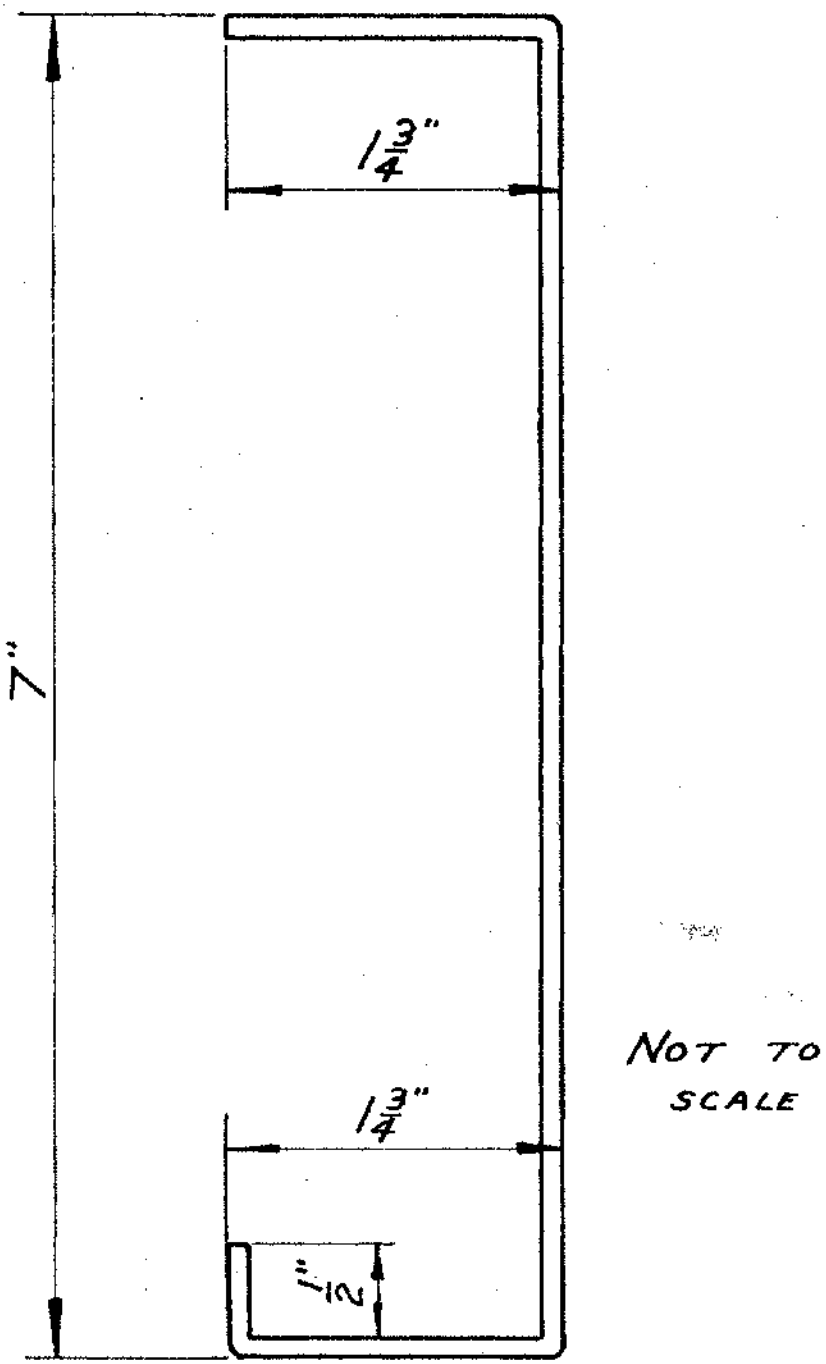
has been curved up slightly at the front end to facilitate coupling of the tractor and trailer.

The cross members of the trailer will be channel shaped and will be bent from low alloy high tensile steel. The shape of these members is shown by the drawing in Table 32 on page 73. It will be noted that a small flange has been turned up from the bottom leg to avoid having a sheared edge on the tension side of the beam. This has been done to prevent the possibilities of fatigue failures resulting from rough edges, as was also true in the case of the upper fifth wheel plate supporting members.

By referring to Table 8 on page 21, it will be seen that the load in a 20 foot trailer is distributed over a length of 204". Dividing the payload of 30,000 pounds by 204" gives a load figure of 147 pounds per lineal inch of trailer length. The figures for the 20 foot trailer are used because the load is most concentrated in this length of trailer and the same concentration may occur in any length trailer. This load figure will be used in calculating the size of the cross members.

The spacing of the cross members depends on the thickness of the floor. It has been found from long experience that a cross member spacing of 24" is the maximum that can be safely used when the floor is made of 1 1/8" thick white oak. This will therefore be used as the criterion of design for the floor and cross members.

TABLE 32. CROSS MEMBERS



MATERIAL: LOW ALLOY HIGH TENSILE STEEL.

A comparison of different floor thicknesses and the resultant cross member spacing will be made to determine which combination is the lightest.

Assuming a 24" spacing of the cross members when 1 1/8" thick oak is used and assuming that the flange width of the cross member is 2 1/2" then the maximum unit stress is found as follows:

M = maximum bending moment
A = pounds of load per lineal inch of floor 1" wide
p = maximum unit stress in pounds psi
I = moment of inertia of section
S = sectional modulus
b = width of section
d = depth of section

$$M = 10.75A\,\frac{(21.5)}{4} - 10.75A\,\frac{(21.5)}{2} = 57.7A$$

$$I = \frac{b\,d^3}{12} = \frac{1\,(1.125)^3}{12} = .119$$

$$p = \frac{M}{S} = \frac{M}{\frac{I}{c}} = \frac{Mc}{I} = \frac{57.7A\,(.563)}{.119} = 273A$$

The cross member spacing for 1" thick oak and a stress of 273A is now found:

L = cross member spacing

$$I = \frac{1\,(1)^3}{12} = .0835$$

$$M = \frac{pI}{c} = 273A\,\frac{(.0835)}{.5} = 45.6A$$

$$M = \frac{L}{2}(A)\frac{(L)}{4} - \frac{L}{2}(A)\frac{(L)}{2} = \frac{A\,L^2}{8}$$

$$L^2 = \frac{8M}{A}$$

$$L = \sqrt{\frac{8M}{A}} = \sqrt{\frac{8\,(45.6A)}{A}} = 19"$$

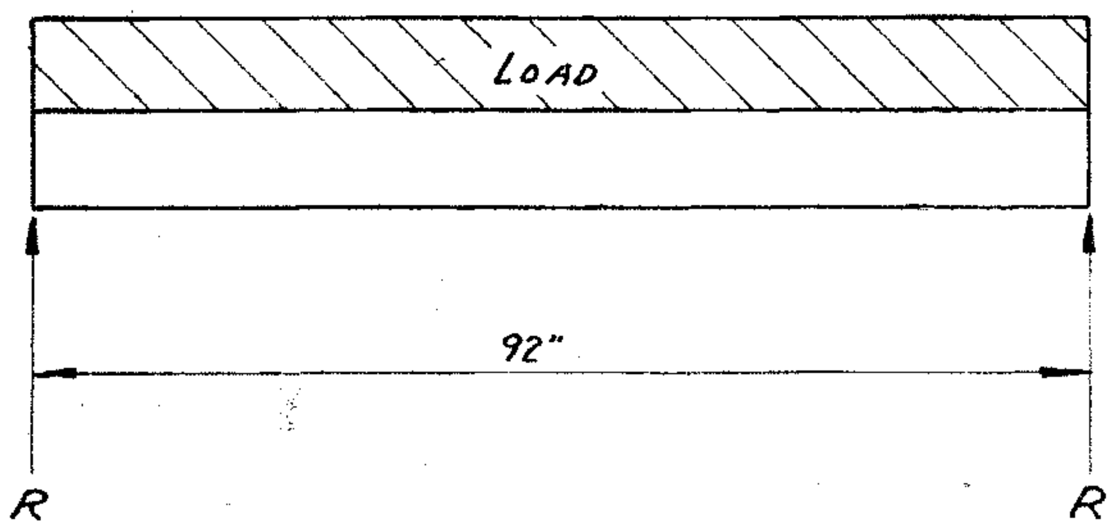
The spacing for 3/4" oak is found to be 14.4" by the same method. These spacings are shown by the Table 34 on page 78.

The load diagram on the cross members is shown by the drawing in Table 33 on page 76. The load is the cross member spacing multiplied by the figure of 147 pounds per lineal inch of trailer length. The endurance limit divided by a factor of safety of 3 gives a maximum allowable unit stress of 15,000 pounds per square inch. This higher factor of safety is needed to take care of any non-uniform distribution of the load and has been found to be about as small as can be safely used. The member is assumed to be the shape shown in Table 32 on page 73 and the small turned-up flange at the bottom will be neglected. A thickness of 9 gauge is the proper thickness for a cross member spacing of 24". This is verified as follows:

9 GA = .1495"

$$I = \frac{(.1495)\ (7)^3}{12} - 2\ (.1495)\ (1.601)\ (3.425)^2$$

TABLE 33. LOAD ON CROSS MEMBERS.



$$I = 9.92$$

$$S = \frac{I}{c} = \frac{9.92}{3.5} = 2.834$$

$$p = \frac{M}{S} = \frac{40,572}{2.834} = 14,316 \text{ lbs. psi.}$$

In a like manner, 12 gauge (.1046") is found to be the requisite thickness for a spacing of 19" and has a unit stress of 15,300 pounds psi. Similarly, 14 gauge (.0747"), is found to be the proper size for a spacing of 14.375" and has a unit stress of 16,590 pounds psi. These latter two cases are stressed slightly over the allowable stress, but are deemed to be satisfactory in view of the large factor of safety. Table 34 on page 78 shows the weights of the various floor thicknesses and cross member combinations in pounds per lineal inch of trailer floor length. This shows that 3/4" floor lumber with a cross member spacing of 14 3/8" is the lightest combination and has been chosen for the trailer. It is not practicable to use a floor thinner than 3/4", so no others were considered.

The load on the landing gear support channels is shown by Table 35 on page 80. This is figured for a 20 foot trailer as this is the worst load condition under which the landing gear must operate. The load on the landing gear is found as follows:

TABLE 34. WEIGHTS OF FLOORS OF DIFFERENT THICKNESSES.

FLOOR THICKNESS	CROSS MEMBER SPACING	WEIGHT			TOTAL WGT. PER LIN. IN. OF FLOOR LGTH.
		WHITE OAK (92" × CROSS MEMBER SPACING)	ONE CROSS MEMBER	TOTAL	
$1\frac{1}{8}$"	24"	62.9 LBS.	41.8 LBS.	104.7 LBS.	4.362 LBS.
1"	19"	44.2 LBS.	29.3 LBS.	73.5 LBS.	3.868 LBS.
$\frac{3}{4}$"	$14\frac{3}{8}$"	25.1 LBS.	21.2 LBS.	46.3 LBS.	3.220 LBS.

Payload = 30,000 pounds
Trailer body weight = 3,000 pounds (estimated)
Total load = 33,000 pounds
R = total load on landing gear

Moments about center line of axle

$$- 12.5\, R_2 + 10.25\, (33,000) = 0$$
$$R_2 = 27,000 \text{ lbs.}$$

The arrangement of the landing gear support channels is shown by the drawing in Table 36 on page 81. The required size of the support channels is found as follows:

M = maximum bending moment
S = required sectional modulus
p = maximum unit stress in pounds per square inch

$$p = \frac{55,000}{1.5} = 36,600 \text{ lbs. psi.}$$

$$M = \frac{(27,000)}{4}\,(11) = 74,300$$

$$S = \frac{M}{p} = \frac{74,300}{36,600} = 2.03 \text{ (Required)}$$

In the study of the floor cross members it was found that a 12 gauge member shaped like the drawing in Table 32 on page 73 has a sectional modulus of 2.10. This size member will therefore be used for the landing gear support channels. The factor of safety of 1.5 seems very low but it must be taken into consideration that the load on these members is static and that all of the load is not transmitted from the side into the ends of the members but that actually quite a bit of the load is transmitted from the portion of the floor directly above the members into the members

TABLE 35. LOAD ON LANDING GEAR.

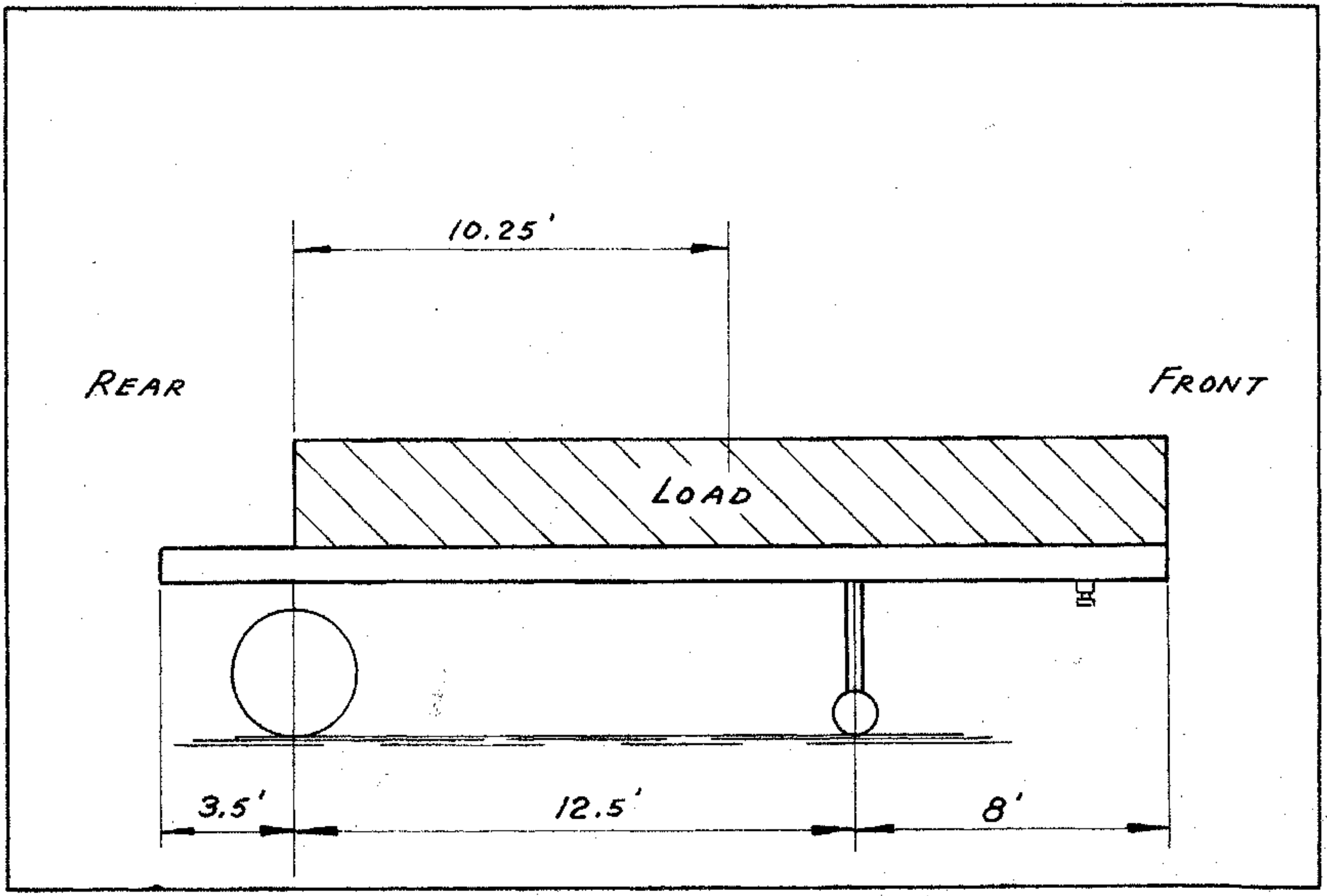
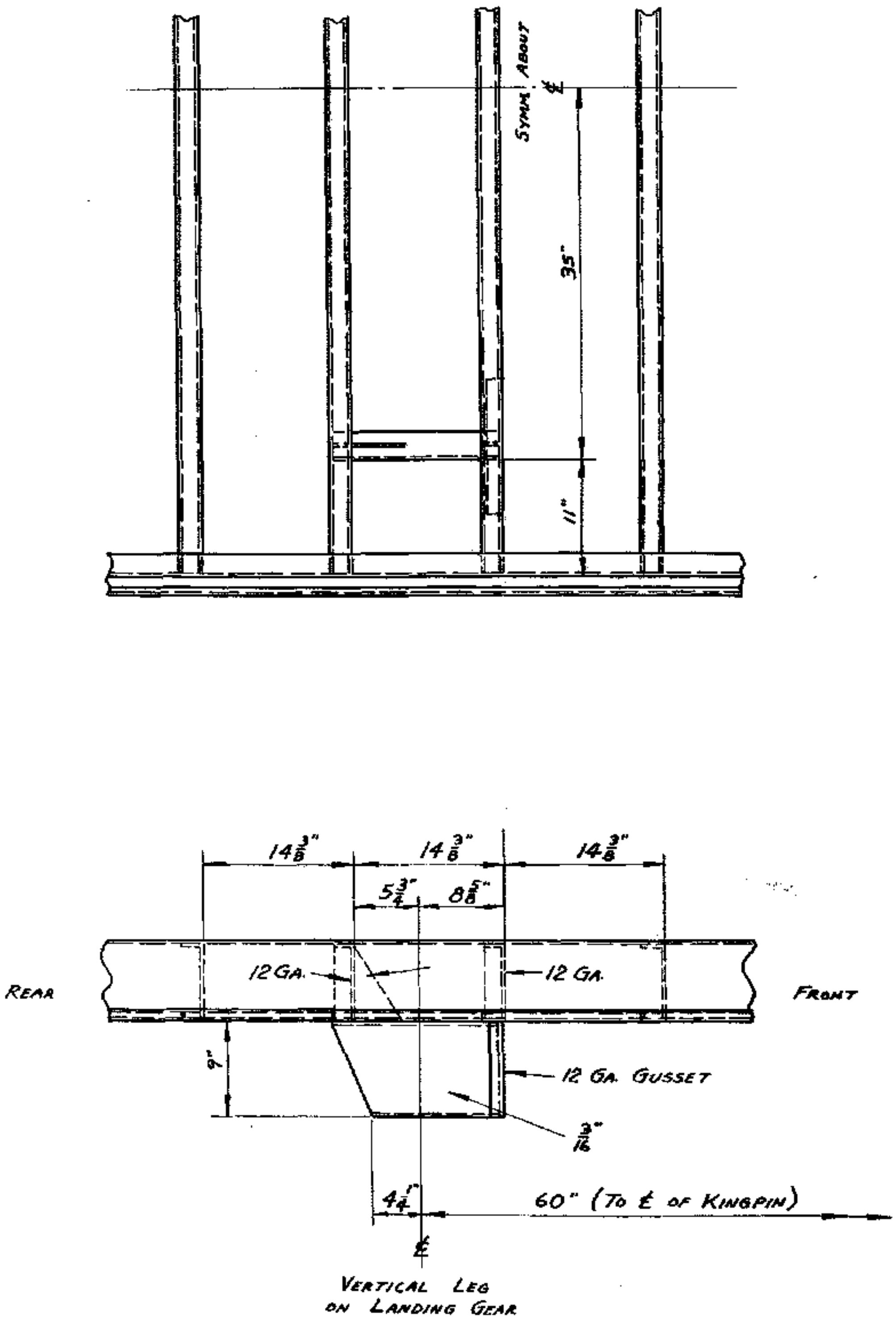


TABLE 36. DETAIL OF LANDING GEAR SUPPORTS.



NOTE:-
ALL MATERIAL LOW ALLOY HIGH TENSILE STEEL.

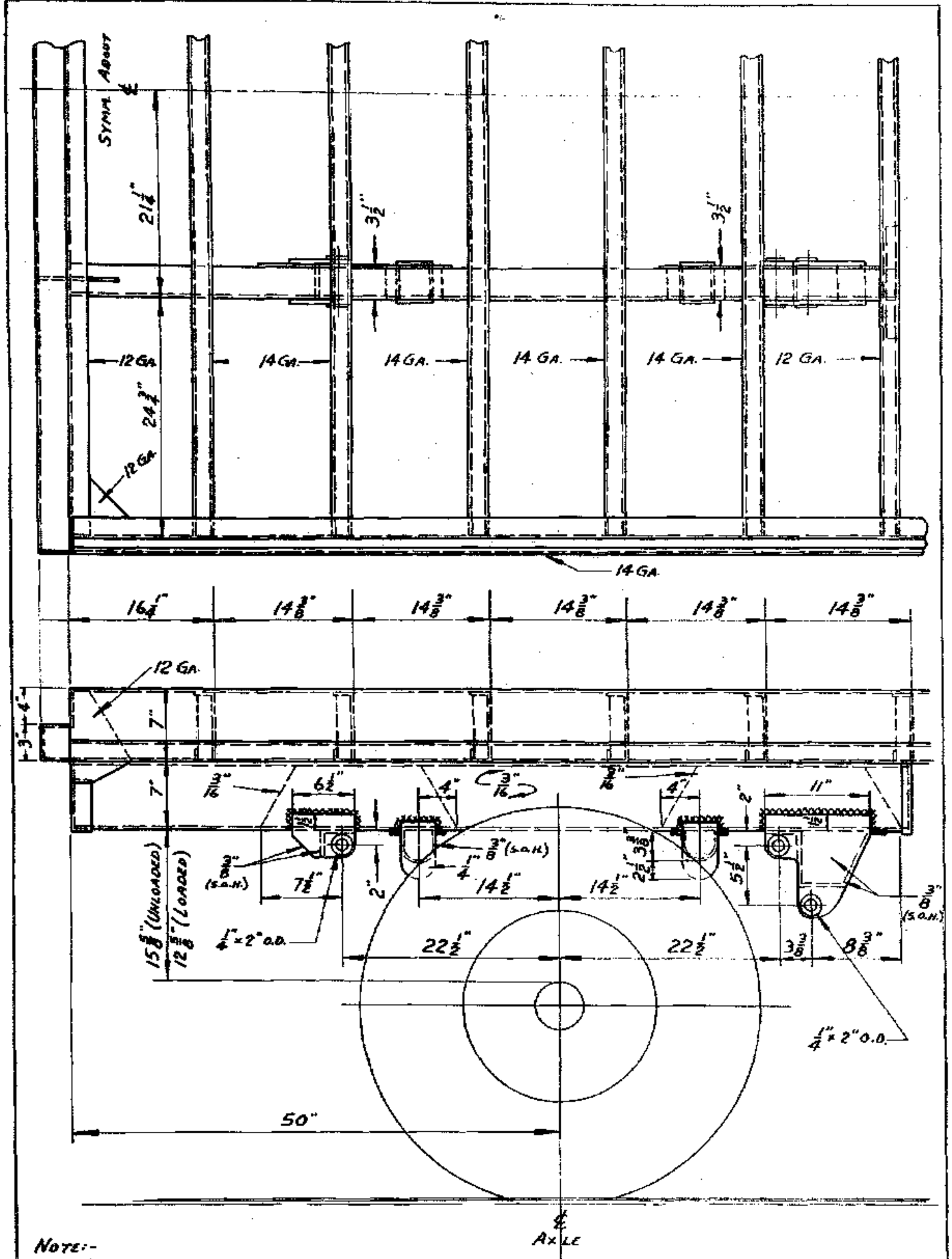
SCALE: $\frac{3}{4}$" = 1'-0"

themselves. This will reduce the calculated bending moment and this was taken into consideration in the selection of the low factor of safety. The members have actually been used in service in the 12 gauge thickness and been found to perform satisfactorily. The short longitudinal channel is made of 3/16" material because it has been found that this is the lightest material that will stand up in this place. The landing gear is bolted on to this member and channels thinner than 3/16" soon get bent up badly. The longitudinal member is braced on the front end by a small 12 gauge gusset as is shown by Table 36 on page 81.

The assembly of the structural members and spring hangers and stops has been shown by the drawing in Table 37 on page 83. The dimensions on the spring assembly as shown give a floor height at the rear end of approximately 52 1/2" when the trailer is empty and approximately 49 1/2" when the trailer is fully loaded. With this floor height the trailer will be nearly level when coupled up to most large size tractors. This is very important from an appearance standpoint. The best appearance is attained when the trailer is nearly level or is slightly lower at the rear end.

The spring hangers and auxiliary spring stops have been designed to be fabricated from sheared and flame cut plate and welded directly onto the longitudinal suspension channel. The spring hangers and stops are of the thinnest

TABLE 37. ASSEMBLY OF STRUCTURAL MEMBERS AND SPRING HANGERS AND STOPS AT REAR END.



NOTE:-
ALL MATERIAL LOW ALLOY HIGH TENSILE STEEL, EXCEPT AS NOTED.

SCALE · ¾" = 1'-0"

possible materials, as has been determined by a long series of experiments on actual vehicles. The spring hangers and auxiliary spring stops are placed directly under the longitudinal suspension member so that the entire load is transmitted in a direct line from the channel to the springs and all overturning moments are thereby avoided. This construction overcomes many of the usual troubles inherent in spring suspensions due to the conventional practice of mounting the springs several inches to the outside of the longitudinal channel and supporting them with cantilever spring hangers. The longitudinal channel and its boxing gussets are both made of 3/16" material. This is another place where long experience has proved that this is the lightest material that can be used for this particular application. These gussets are tapered at both ends to avoid stress concentrations caused by too sudden changes of section of the member. The spring hangers and auxiliary stops are welded on and care should be taken to see that they are not welded across the ends which would be across the direction of stress. This might cause a fatigue failure at the weld. The manner in which the springs are mounted is shown by Table 21 on page 47.

The sides of the frameless trailer carry the entire load and must be designed to carry this load without buckling. Some frameless trailers use a trussed side made of hollow tubing and cover this on the outside with sheet metal. It is desired to use stressed skin construction in the being designed so that the trailer will be simple to construct and light as possible. Since sheet metal sides have to be used to cover the outside, it is only logical to join them together and provide enough stiffeners so that the side sheets themselves will carry the load. Some trailers have been built using side sheets with vertical corrugations (usually called ribs) made like the drawing in Table 38 on page 86. This type of side has proved to be unsatisfactory because a blow from the inside of the trailer such as might be caused by cargo shifting will deform the rib to a shape somewhat like that shown by the dotted lines in Table 38. This results in the side bulging toward the outside of the trailer. Some trailers have been observed by the writer when the sides have been bulged out as far as six inches. The side sheet construction shown in the drawing in Table 39 on page 87 overcomes the difficulty described above. After the sheets are spot welded in the manner shown on the drawing the side is a very strong one as each rib is then a hollow tube integral with the side. The determination of the thickness of the side sheets and the spacing of the ribs will now be carried out.

Payload = 30,000 pounds
Load on one side = 15,000 pounds
Maximum shear at ends = 7500 pounds
Panel length = 78" to 87"
Rib spacing = 11"
Effective width of panel = 11" - 2 (1 1/8") = 8 3/4"*

* See Table 39 on page 87

TABLE 38. RIBBED SIDE OR ROOF SECTION.

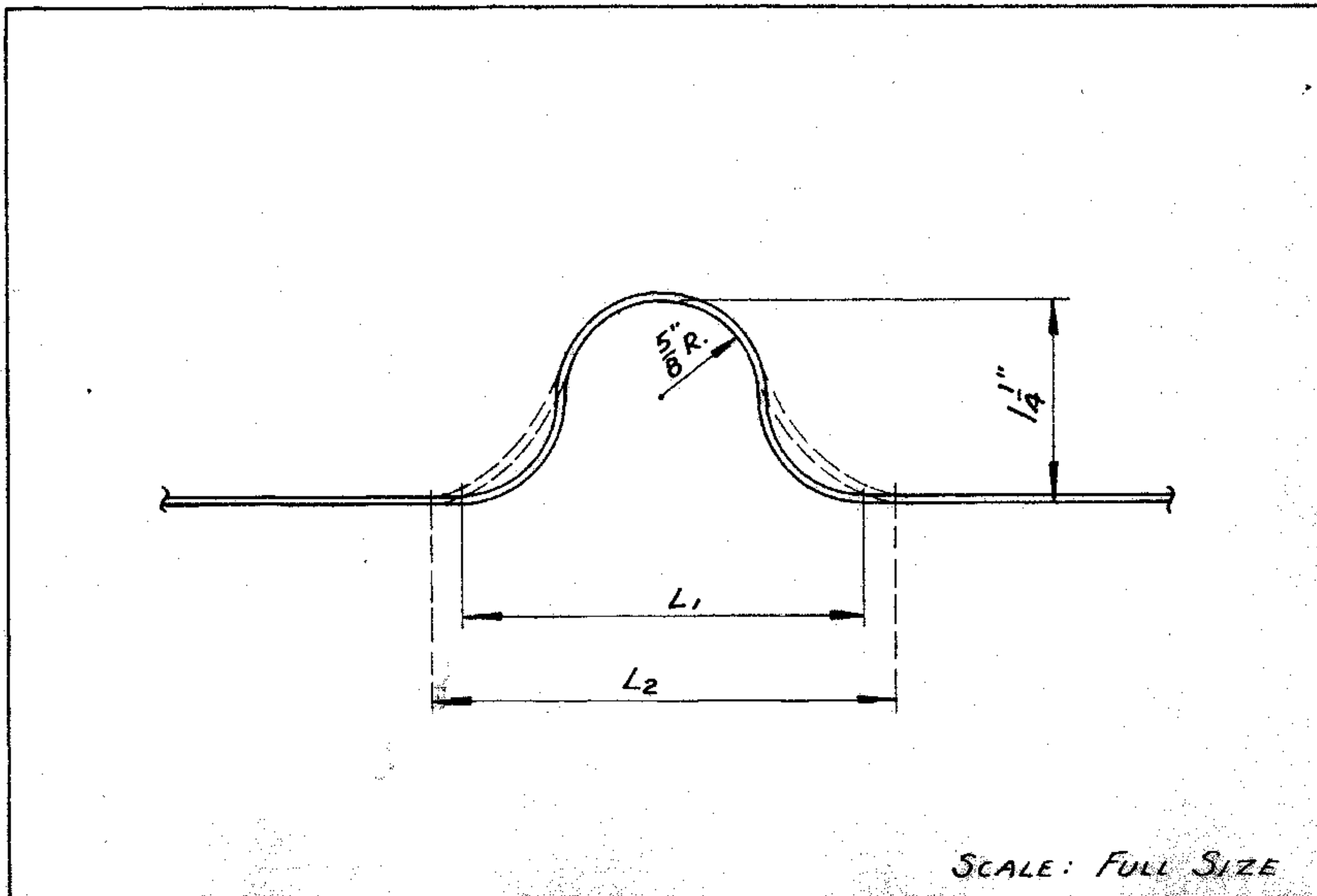
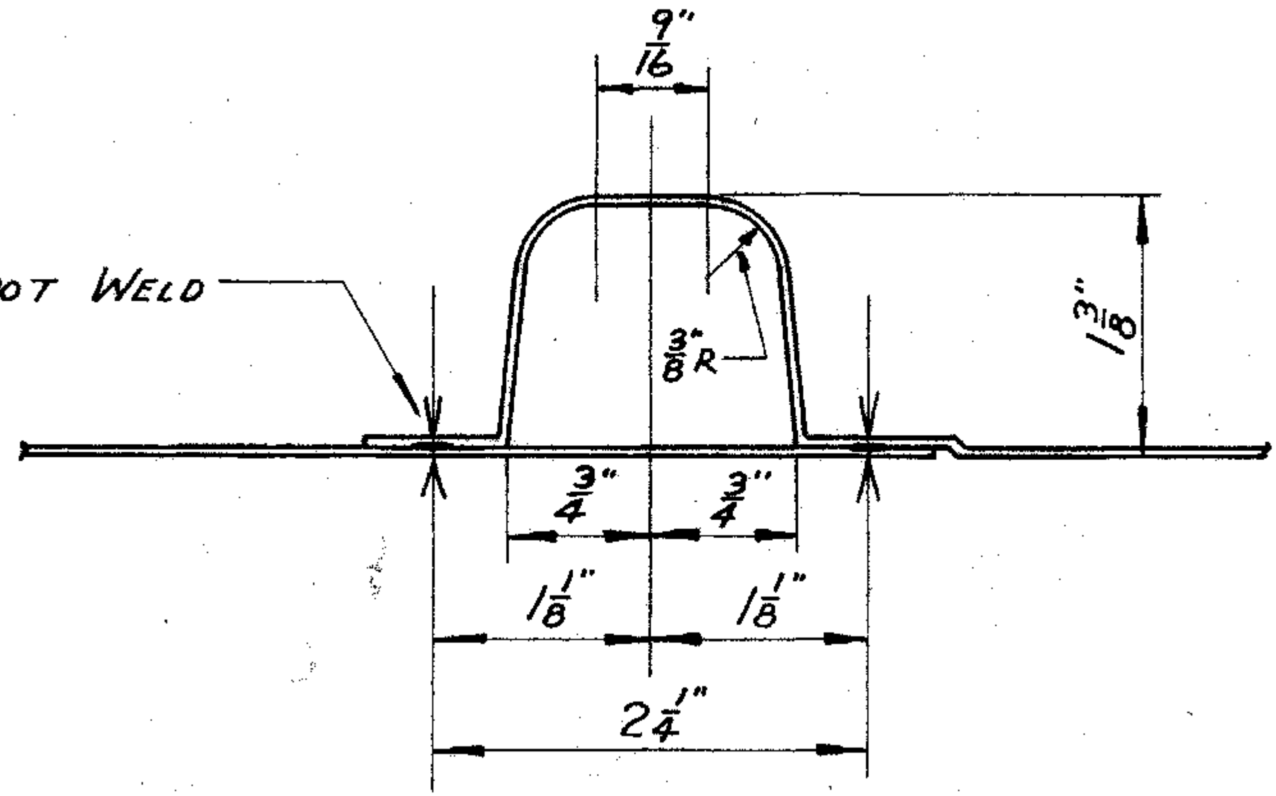


TABLE 39. CROSS-SECTION OF SIDE SHEETS.



SCALE: FULL SIZE

Shear stress per lin. inch = $\frac{7500}{78}$ = 96.2 (for the worst case)

As the critical shear stress for buckling occurs at about 75 per cent of the theoretical value, then a factor of safety of 1.33 should be used.

Critical shear stress = 96.2 (1.33) = 128.2 lbs. per inch

The buckling strength is now found by using the chart on page 5 of "Designing for High Tensile Steels", by H. M. Priest, reprinted from "Railway Mechanical Engineer".

B = effective width of panel
A = length of panel
K = constant from the chart, dependent upon B/A
S_{cr} = critical shear stress for buckling

$$\frac{B}{A} = \frac{8.75}{87} = .1008$$

$$K = 5.4$$

From chart S_{cr} = 125 for panel 20 ga. thick

S_{cr} must not exceed shearing yield point times thickness of the panel.

$$33{,}000 \times .0359 = 1185.$$

Since the critical shearing stress for the 20 gauge panel is practically the same as the critical stress per lineal inch of the sheet as previously calculated and since the critical shearing stress does not exceed that calculated immediately above, the 20 gauge sheet is satisfactory in this application.

TABLE 40. RIB SPACING ON SIDE SHEETS.

THICKNESS OF SIDE SHEETS		REQUIRED RIB SPACING	WEIGHT OF SIDE SECTION ONE FT. HIGH PER LIN. INCH OF TRAILER LENGTH
MFG'S. STD. GA.	INCH EQUIVALENT		
18 GA.	.0478"	13"	.235 LBS.
20 GA.	.0359"	11"	.184 LBS.

The buckling stress of an 18 gauge panel was investigated similarly and it was found that an 18 gauge panel is satisfactory with a rib spacing of 13". In both of these cases, buckling from shear will begin at about the rated payload of the vehicle, but failure will not occur until the stress is increased a very large amount. This buckling has been observed by the writer in actual tests of such panels on complete trailers, but there has never been an instance of complete failure found yet.

Table 40 on page 89 shows a comparison of the two side sheet thicknesses described above. It will be seen that the 20 gauge sheets with 11" rib spacing are much lighter than the 18 gauge sheets with 13" rib spacing. Thinner sheets were not considered as 20 gauge is about as light as can be handled in most shops so that the completed trailer does not show wrinkles in the sides. The 20 gauge sheets will therefore be used in the trailer.

The side sheets should be spot welded together with the spot welds spaced on about 1 3/8" centers. This is done for the sake of appearance, as such a spacing gives considerably more strength than is required to handle the load. However, if the spots are placed much further apart, the sheets tend to buckle and show an opening in the seam between the welds. The factor of safety of the spot welding is calculated as follows:

Spacing of spot welds = 1 3/8"
Length of panel = 78"

$$\text{Number of welds} = \frac{78}{1.375} + 1 = 57$$

Diameter of electrode = 3/16"
Take dia. of spot weld = 1/8"
Shearing strength of spot welds = 45,000 pounds psi*

$$\text{Area of spot} = 3.1416\ (1/16")^2$$
$$= .00815 \text{ sq. in.}$$
$$\text{Total strength of welds} = (57)\ (.00815)\ (45{,}000)$$
$$= 20{,}900 \text{ lbs.}$$

$$\text{Factor of safety} = \frac{20{,}900}{7{,}500} = 2.79$$

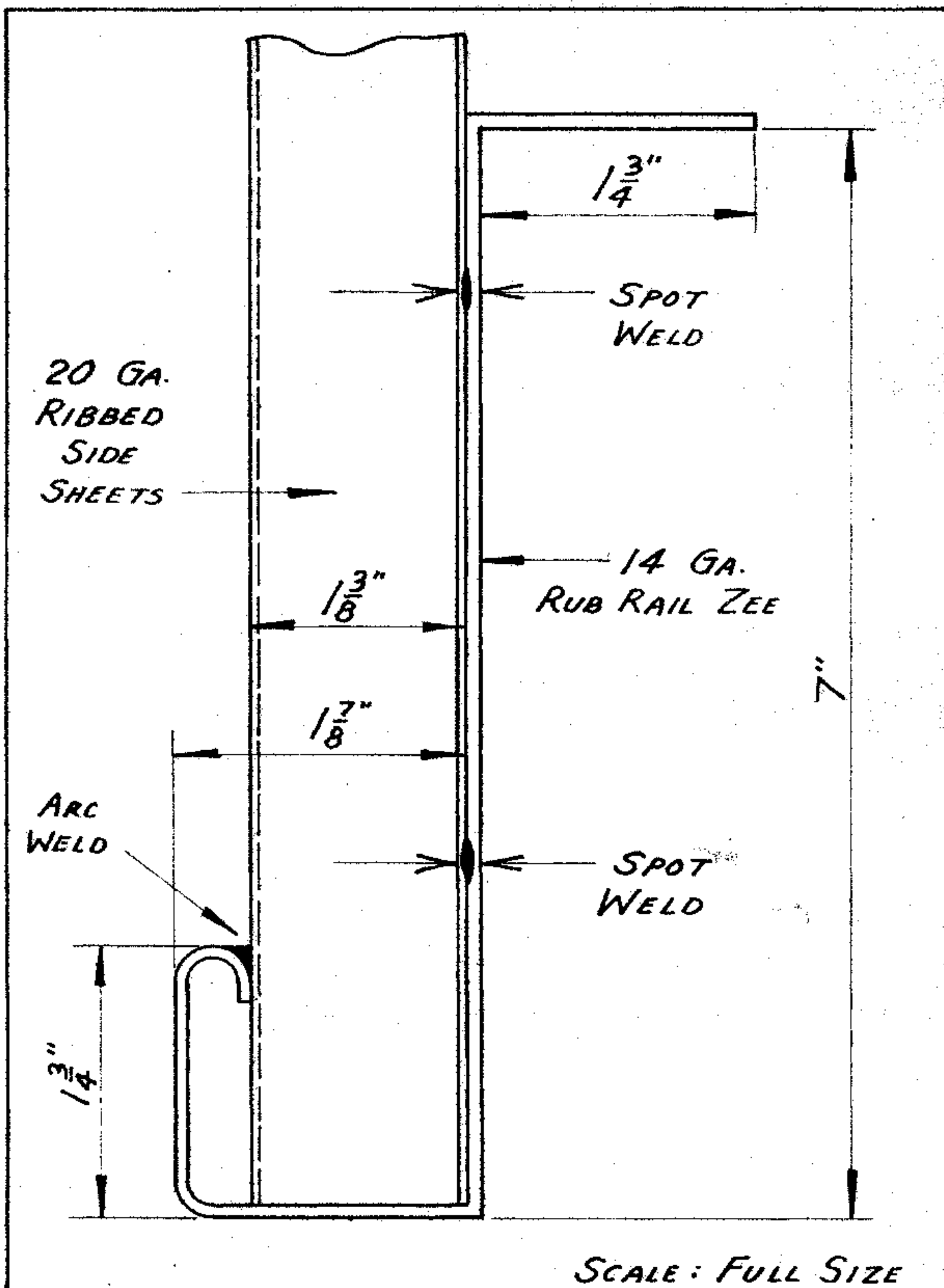
In view of the fact that the strength of spot welds can be affected by such things as variable voltage and varying size of the electrode tips, this factor of safety seems to be about right.

Table 41 on page 92 shows the detail of the assembly of the rub rail zee and the sides. The sides are spot welded to the zees with welds on 1 3/8" centers. The zee and rubrail are formed from one piece and are made of 14 gauge high tensile steel as experience has shown that is about the lightest that can be used here.

The interior of the trailer should be lined with plywood to a height of at least four feet above the floor to prevent the sides of the trailer from becoming bent by hand trucks, boxes or crates accidently striking against them during the process of loading or unloading. This

* From "American Welding Society Handbook, 1942 Edition".

TABLE 41. ASSEMBLY OF RUB RAIL ZEE AND SIDES.

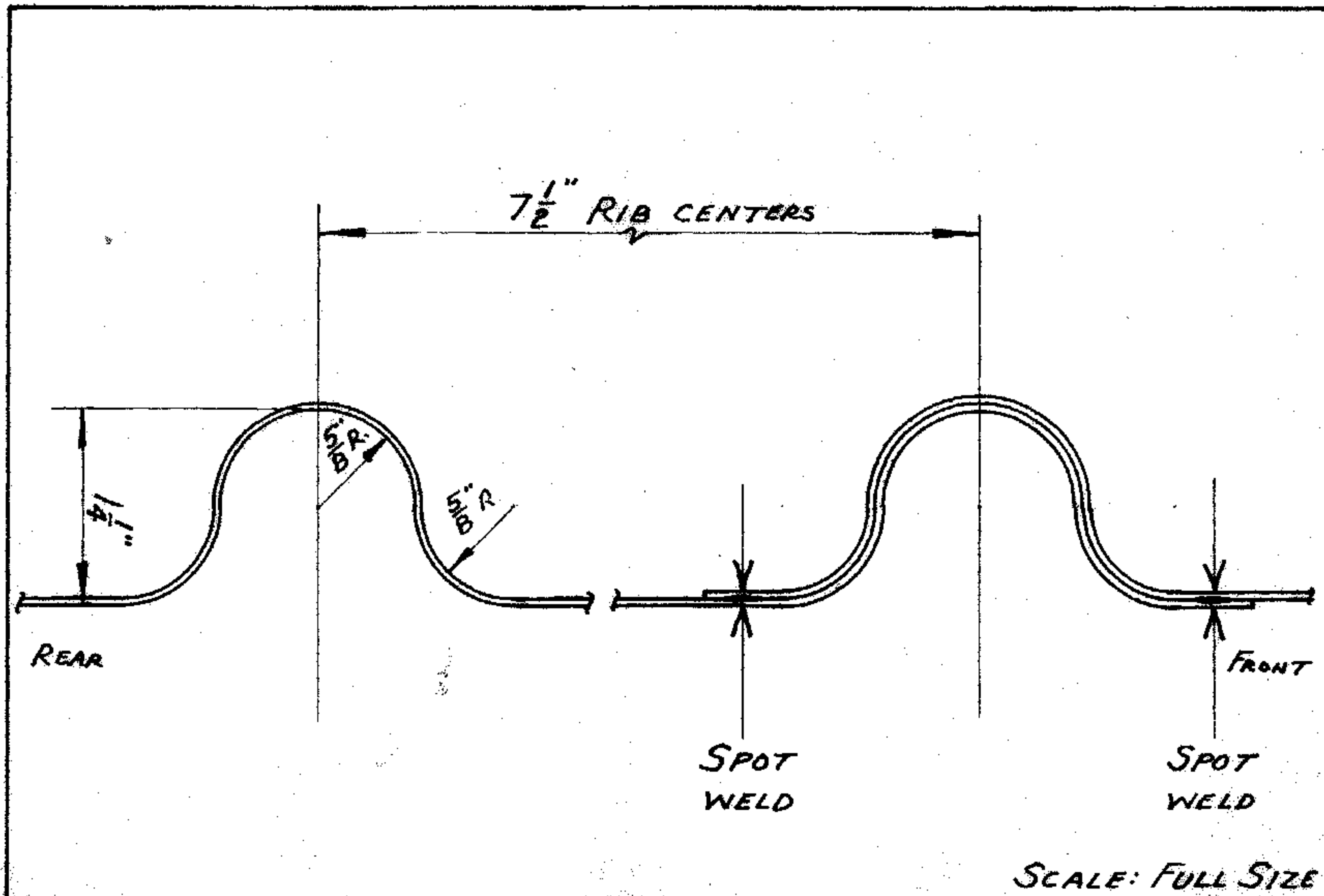


plywood will be satisfactory if only 1/8" thick. It should be held in place by self-tapping sheet metal screws, preferably of the flat countersunk washer head type. As in the case of the floor bolts, the screws should have Phillips recessed heads to facilitate driving with air driven screwdrivers. The screws should pass through and into the hollow column section of the sides, so that they will not protrude to the outside.

The roof of the trailer must not leak and should be strong enough to support a man walking on it, as it is often necessary to climb onto the roof to clean, paint, and repair damaged places. The roof of a trailer is often struck by small overhanging limbs of trees and consequently damaged. When a trailer body twists as the tractor and trailer move over uneven roads, there is a tremendous strain exerted between the diagonally opposite corners of the roof and this will split the roof apart unless provision is made for it to stretch and thus relieve the strain.

There are many types of joints used on trailer roofs, including coin pressed lock joints, single lock joints, lap joints, and soldered seam joints. The most inexpensive and satisfactory of these is the simple lap joint modified so that the lap is made over a pressed rib. This is illustrated by the drawing in Table 42 on page 94. The lap joints are spot welded as shown in the drawing and the

TABLE 42. ROOF CONSTRUCTION



spot welds are made on 1 3/8" centers so that the sheets will fit together snugly all along the seam and prevent leaks.

The roof made out of wide sheets is used also because it is of one-piece construction and therefore does not have any parts to work and rub against each other. The ribs are placed on 7 1/2" centers because this rib spacing is strong enough to hold up a heavy man, because the panels have a very narrow width and do not rumble and vibrate, and because this spacing works out right for the use of a 60" sheet width without any necessary shearing waste. The design of the roof is made purely on a basis of what experience has shown to be the proper sizes and construction to do the job properly. The ribbed roof section also has the advantage that it will deform and relieve the diagonal strains discussed above. This was illustrated in the drawing in Table 38 on page 86. The assembly of the roof channel where the roof welds on to the trailer is illustrated by the drawing in Table 43 on page 96. The roof channel parts are formed from 14 gauge low alloy high tensile steel and are welded together as shown. The outer part is welded to each rib of the sides and the channel is spot welded to the sides with spot welds spaced on 1 3/8" centers. The roof is welded on with a continuous arc weld, making a waterproof joint.

TABLE 43. ROOF CHANNEL ASSEMBLY.

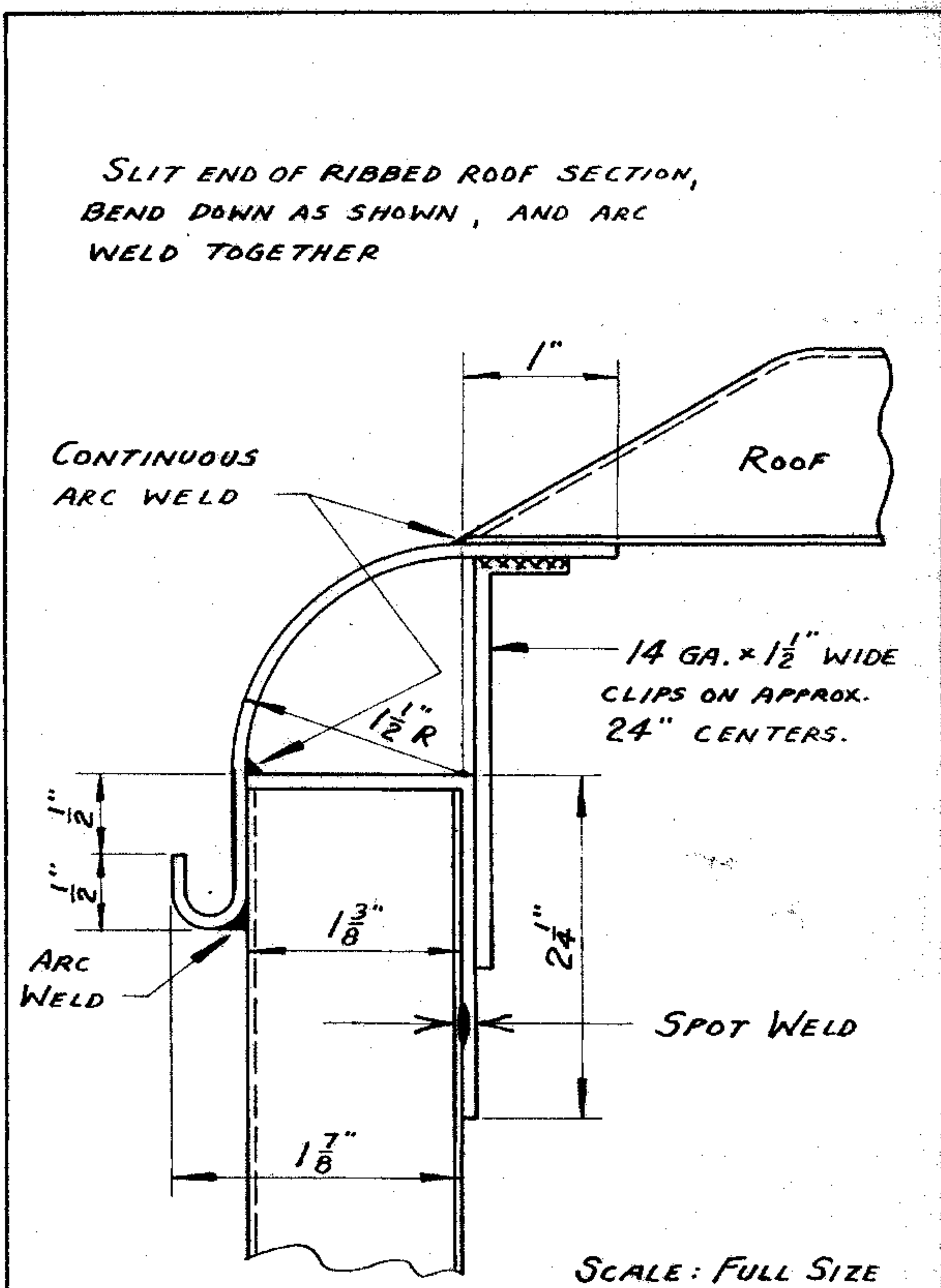


TABLE 44. DOOR MEMBERS.

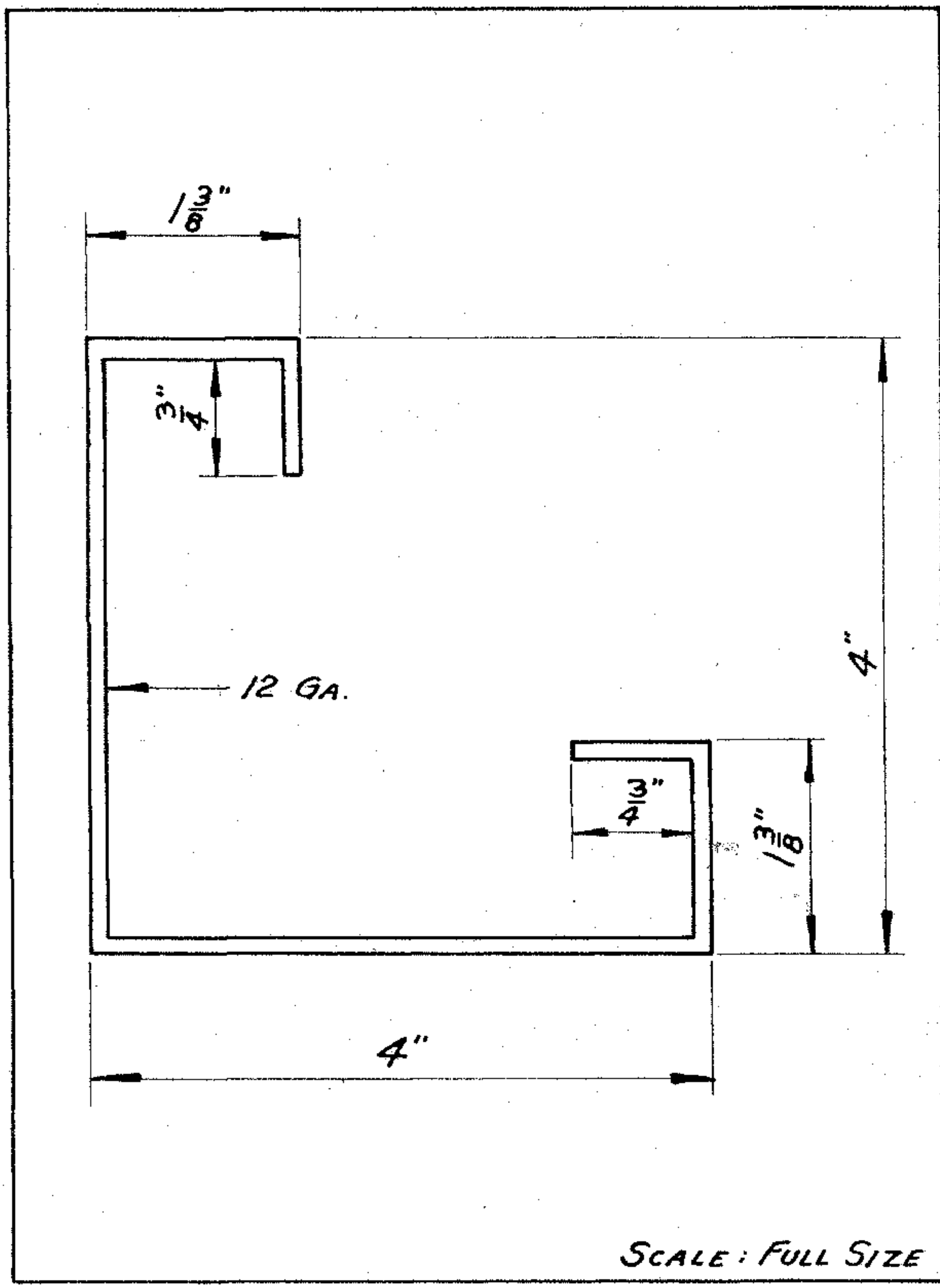
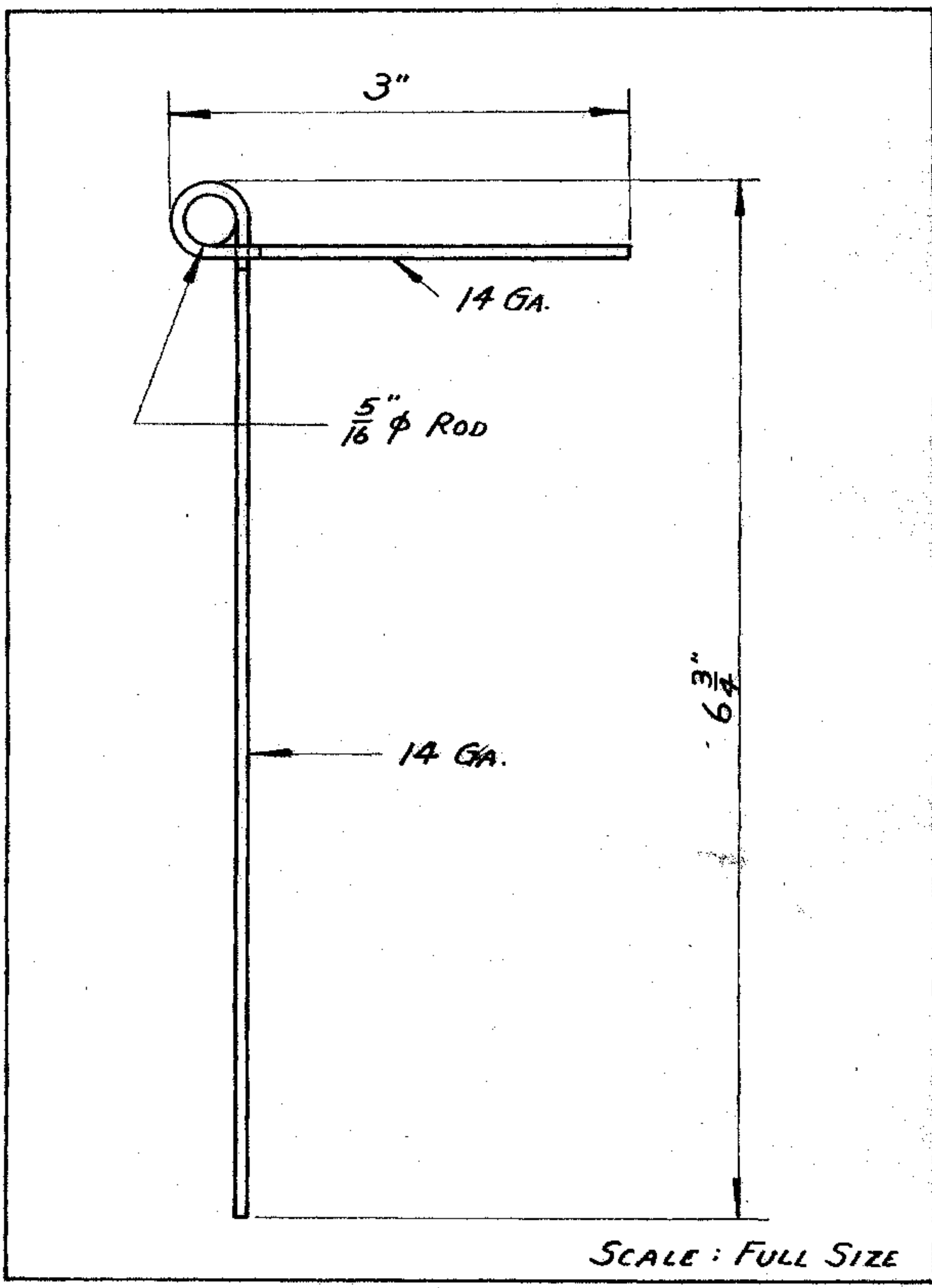


TABLE 45. CONTINUOUS DOOR HINGE.



The rear end of the trailer is framed in by one horizontal and two vertical door members illustrated by the drawing in Table 44 on page 97. The members are of 12 gauge high tensile steel and are as light as they can be made and yet withstand the strains on the back end of the trailer, as has been determined by experience. The members are welded together at the top of the trailer and the vertical members are welded into the under structure at the bottom.

The conventional method of mounting the rear doors on the trailer is to use three flat hinges on each door. Table 45 on page 98 shows the continuous hinge designed for the trailer. This hinge is made of the proper dimensions so that it can be spot welded to the rear door and bolted onto the outer part of the vertical door members. The bolts should be 5/16" diameter and spaced about 4" apart for good appearance. The short 3" leg bolts onto the door frame member, while the long leg of the hinge spot welds onto the door. The greatest advantages of the continuous type of hinge are that it effectively seals the edges of the door against water seepage, that it braces the door along the entire edge, and that it is cheaper and lighter than the regular strap type hinges.

The design of the door lock is shown by the drawing in Table 46 on page 100. This is a cam action lock made up of steel castings and soft open hearth steel pieces welded

TABLE 46. DOOR LOCK.

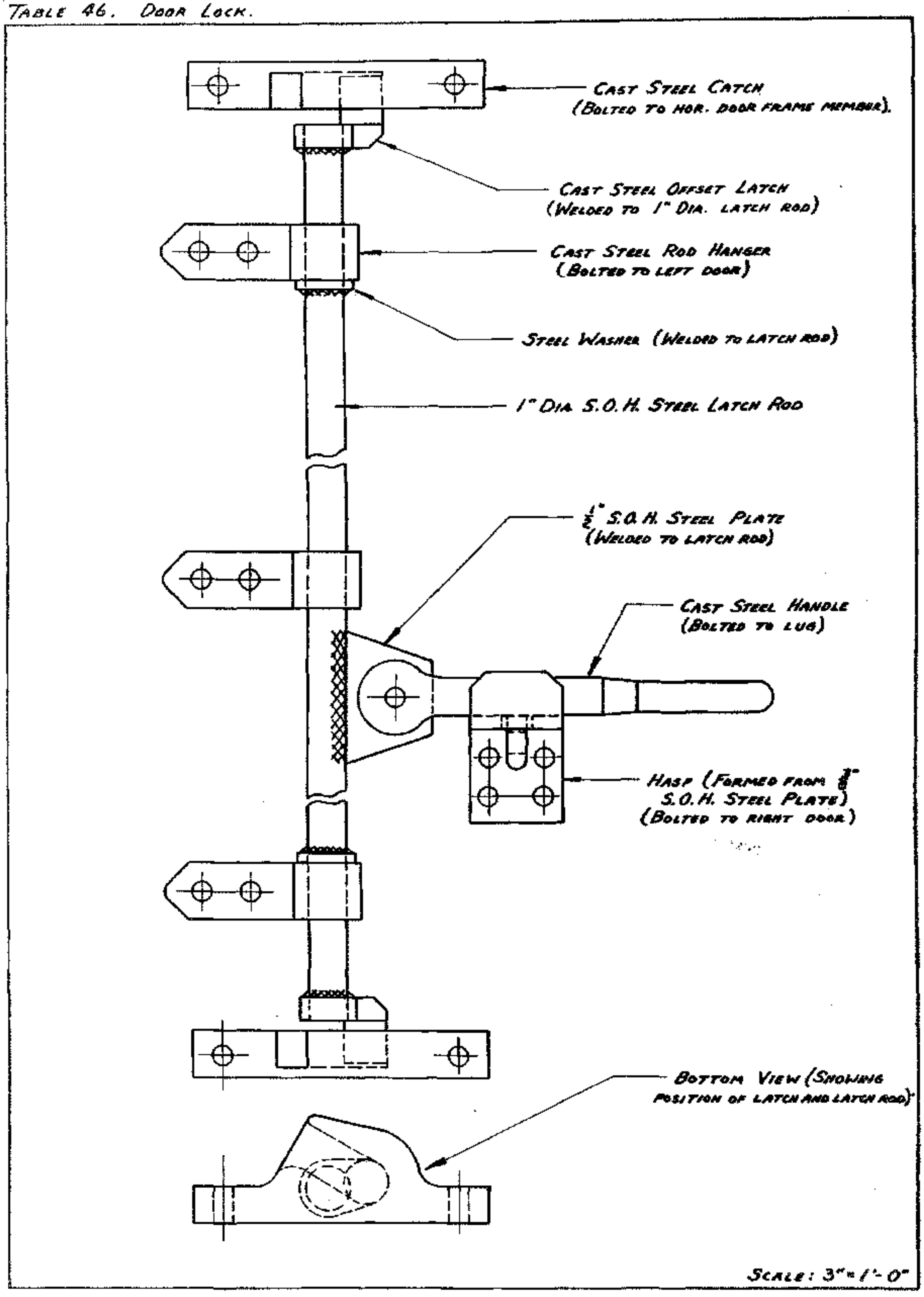
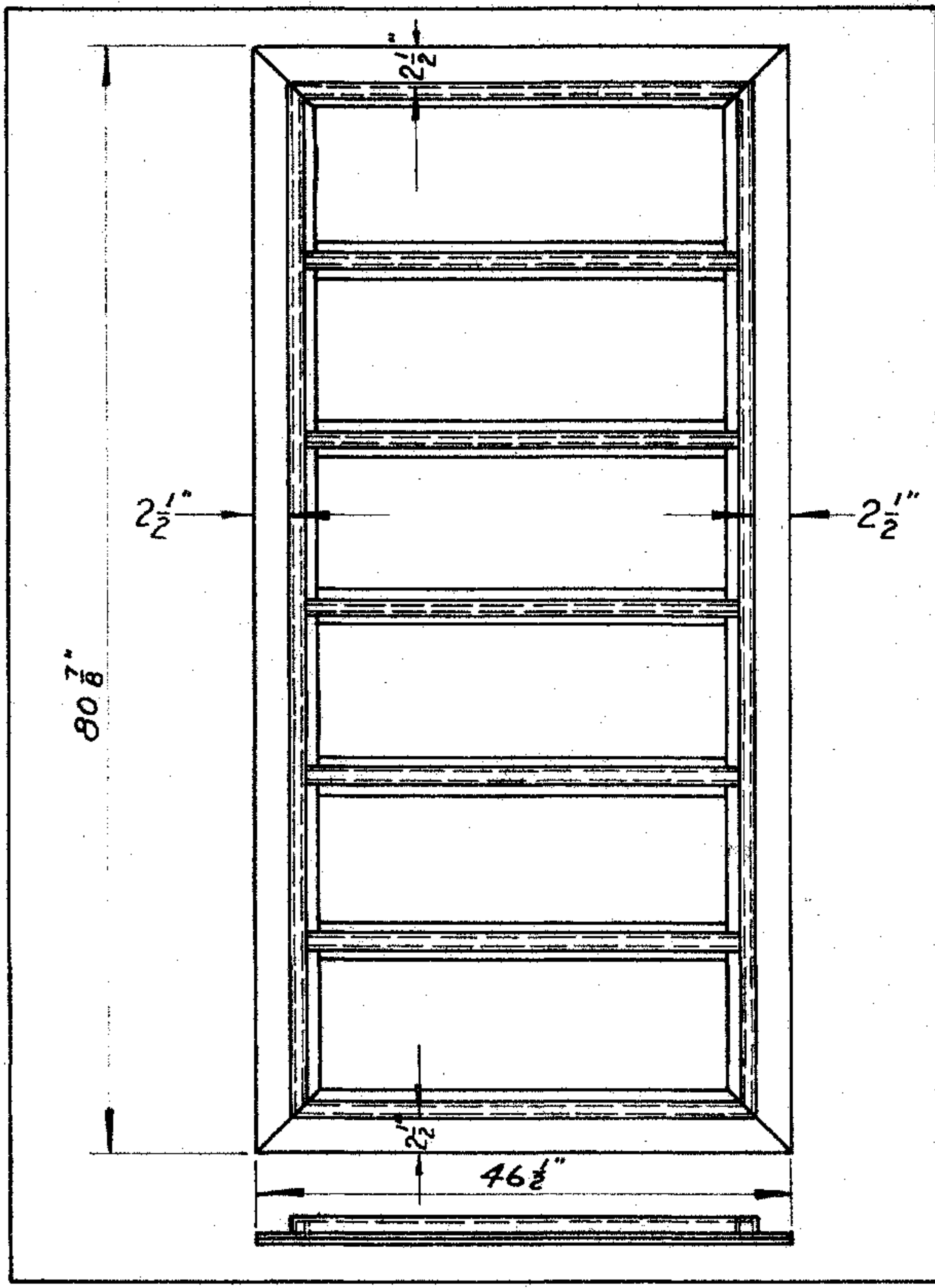


TABLE 47. ASSEMBLY OF REAR DOORS.

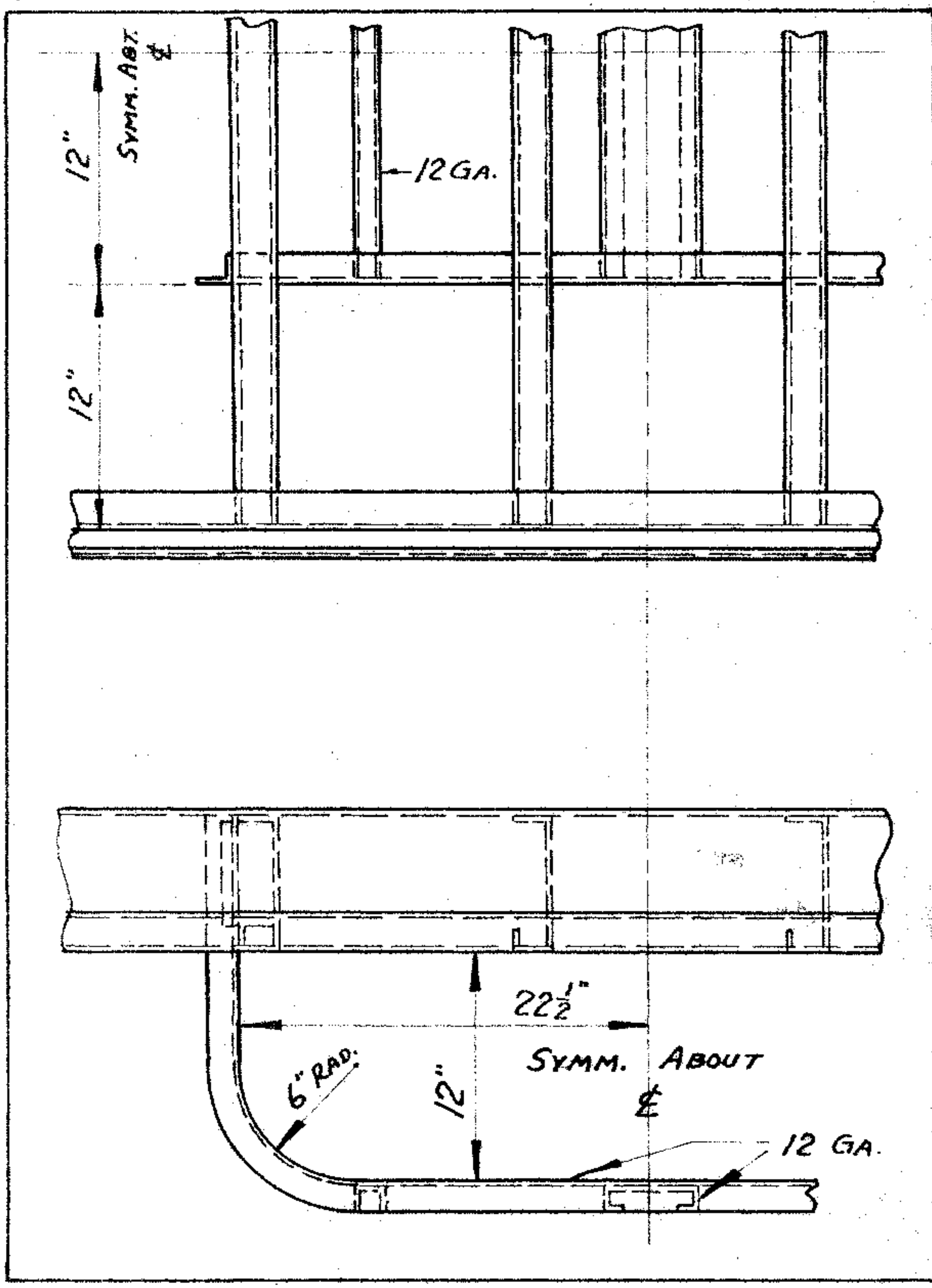


together as shown in the drawing. Every effort has been made to make this door lock as strong as possible without being too heavy, as the door lock of a trailer receives very severe service and most locks will not work properly after a few months use. This lock was developed after trying four different locks made by various manufacturers and finding that all were too light to stand up in commercial freight service. Because of the cam action, the lock illustrated will engage when the door lacks three inches of being closed, so that it can even be used to shut doors that have become badly bent.

The rear doors are shown by the drawing in Table 47 on page 101. They are made from 20 gauge high tensile steel and are braced by ribs pressed from the same material and spot welded on. The two doors fit with the ribs toward the inside of the trailer and overlap 1 1/2" at the center when closed. This construction has proved very satisfactory and is very light. The bracing ribs provide a good place to use small scrap pieces of sheet.

The trailer needs a bumper on the rear end to protect the door structure when the trailer is backed up to a loading platform, since most drivers simply back up until the trailer strikes against the platform. The rear cross member is formed so that the bumper is an integral part of the cross member. This provides a strong bumper across the entire rear end of the trailer, with the

TABLE 48. SPARE TIRE CARRIER.



addition of only about twenty extra pounds to the trailer weight. The bumper is braced with a gusset at each end and at each longitudinal suspension channel. Experience has proved that 12 gauge high tensile steel is the lightest material that can be used for this application.

The spare tire carrier is a simple welded structure mounted underneath the trailer and arranged to carry one spare tire. The tire carrier is fabricated from 12 gauge high tensile steel. The tire is held in by a short angle member that fits the contour of the rim and is bolted to the center member of the tire carrier. The spare tire carrier is shown by the drawing in Table 48 on page 103. It is arc welded to two of the cross members. The tire carrier is another part of the trailer that is designed on the basis of previous experience.

ASSEMBLY OF COMPONENT PARTS

The assembly work on the trailer should be done on an assembly line to obtain the many savings inherent in this type of layout. It is not necessary to move the assembly line continuously, as all the units can be moved progressively from one station to another as each operation is completed.

First, the under structure should be welded together in a jig, in the bottom side up position. After the welding is completed, the spring assembly and axle should be mounted and the unit then turned over. The door frame members are next arc welded in place. The side sheets are welded together on the stationary welding machine next. They are then assembled on the trailer with each side in one piece and the front also in one piece. The roof channels are next assembled on a jig and then transferred to the trailer, after first welding together. The sides and roof frame are now clamped in place and the whole assembly welded together with portable spot welders. After this the roof is spot welded together into one piece, transferred to the trailer, and then arc welded into place. Next the wood floors and plywood lining are installed. The rear doors are attached next. Finally, after all welding operations have been completed, the landing gear, wheels, rims, and tires are mounted. The lights and wiring can be installed at the same time. The trailer is now ready for the paint shop.

PAINTING

Painting is an operation in the manufacturing process that must be performed very carefully as the appearance of the vehicle can be either enhanced or spoiled by the paint job. Painting consists of only four simple operations and each will be discussed briefly.

Cleaning should be very thorough as the adherence of the paint depends upon how well it is accomplished. The easiest way to clean the vehicle is to use one of the many commercial preparations made for this purpose.

Sealing of the wood surfaces has already been discussed, as this should be done before the wood is assembled. A suitable primer should now be applied with a spray gun, taking care to see that all surfaces are entirely covered.

Sanding is necessary with most primers to secure a smooth final coat. However, it will be well to use one of the recently developed primers that does not require much sanding, as this is a slow, hand job.

Painting should be done by the most expert spray gun operator available, as this operation is one which shows every little flaw. If properly done and if the modern synthetic enamels are used, the resulting surface will be a glistening coat that will give years of service.

THE COMPLETED UNIT

The completed trailer will have the following dimensions:

Over-all length	= 30' 4"
Over-all width	= 95 3/4"
Over-all height	= 138" (unloaded)
Inside length	= 28' 10"
Inside width	= 91 3/4"
Inside Height	= 84"
Floor height	= 52 1/2"

The weight of the complete trailer equipped with vertical landing gear, air brakes, spoke wheels, and size 10.00 x 20 tires has been estimated to be 6,586 pounds. This is a very low weight for this size trailer and such a weight would meet with quick approval from any trailer user.

Overhead costs and administrative expenses vary so much with different plants that the cost figures will be presented as out-of-pocket costs only and will not include any burden or taxes. The cost estimated on this basis is as follows:

Material cost =	$856.71	
5% waste =	15.88	
Total material =	872.59	
Labor =	320.00	(400 hours @ .80 hour)
Total cost =	$1,192.59	

PRELOADING FOR PREVENTION OF FATIGUE FAILURES

In welding up a structure such as the under-structure of this trailer, stresses due to the heat caused by arc welding are likely to lock up stresses into the structure. It would be possible to relieve these stresses by annealing, but this is not possible in most shops where a structure of this size is concerned. An alternative is to load the unit about fifty per cent beyond its normal payload of 30,000 pounds. This will not cause a permanent set in any of the parts unless they have had an additional stress placed on them by the locked-up stresses previously mentioned. In this case the yield point might be exceeded, in which case the particular member would take a small permanent set and thereby relieve the locked-up stress. Since the S-N curve for steel has a sharp "knee", such a slight reduction in subsequent stresses on the member might prolong its fatigue life many times.

NORMAL MAINTENANCE

The moving parts of a trailer are comparatively few but they require regular maintenance to assure satisfactory operation. A brief maintenance outline is presented below:

1,000 Mile Check

1. Check tires.
2. Lubricate spring assembly.
3. Check lights.
4. Check brakes.

5,000 Mile Check (In addition to the above)

1. Remove wheels and repack bearings.
2. Check grease seals on axle.
3. Check brake lining and brake drums.
4. Adjust wheel bearings.
5. Adjust brakes.
6. Check axle alignment

Six Months Check (In addition to the above)

1. Check over entire trailer for any necessary repairs.
2. Touch up all paint where any rust shows.
3. Tighten any loose bolts or screws.

Preventative maintenance such as outlined above will prolong the life of the vehicle a great deal and will help prevent costly break-downs on the road.

CONCLUSION

This Thesis has presented the various factors affecting the design of highway freight trailers and has listed the materials available at the present time for use in building them. The various standard automotive components have been studied and charts compiled so that the lightest and most inexpensive parts could be selected for use in the vehicle.

The structural parts of the trailer have been designed so that they can be fabricated and assembled with a minimum of tools and expense. As many of the members of the trailer had to be worked out from a basis of past experience due to lack of exact knowledge of the impact loads on them, the design could be further refined by making a study of the loads on these members by using electrical strain gauges. The design of a number of the parts would also benefit from a study of the strain distribution in them by means of the Stresscoat method.

It has been endeavored to design a lighter, stronger, better, and cheaper trailer and the final design represents a large saving in weight and consequently also in material costs over previous designs. The data presented should greatly facilitate working out any future designs.